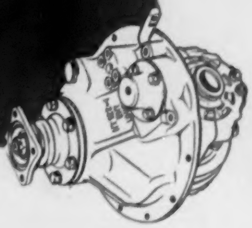
AUTOMOBILE ENGINEER

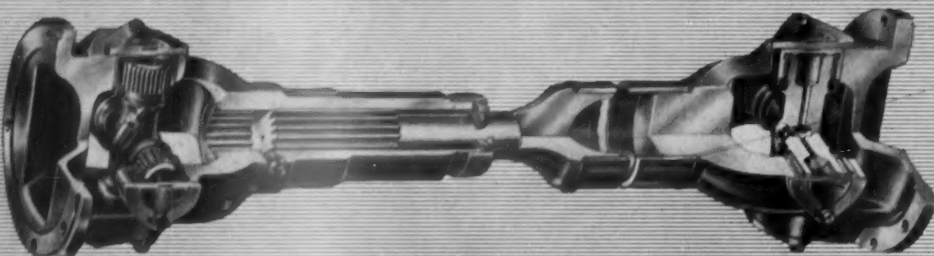
DESIGN · PRODUCTION · MATERIALS

Vol. 45 No. 3 MARCH 1955 PRICE: 3s. 6d.

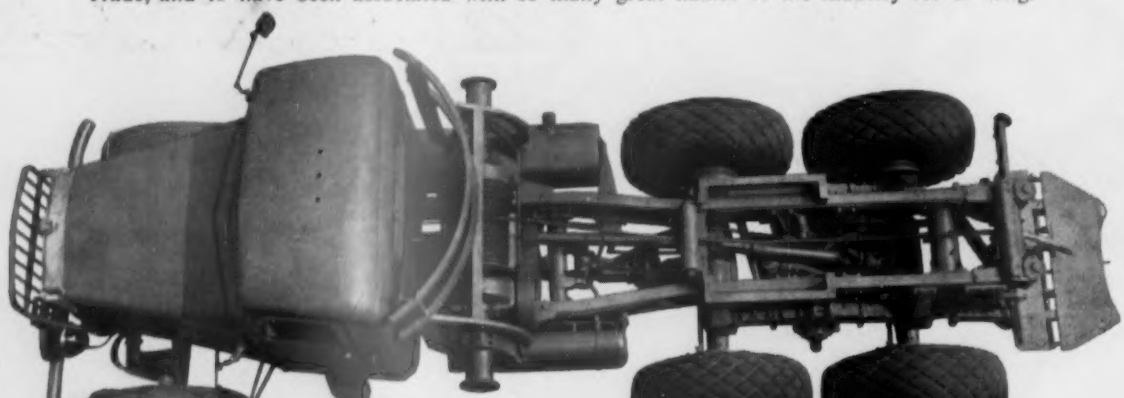


Scammell 8-ton chain drive solid tyre 4 wheeler. Built 1928.
(Hardy Spicer Propeller shaft between gearbox and chain drive axle.)

HARDY SPICER KEEP THE WHEELS TURNING



Scammell lorries were fitted with Hardy Spicer equipment as long ago as 1927. It is an expression of their confidence in Hardy Spicer equipment that Scammell's continue to use it right up to the present day. Hardy Spicer are justly proud of their record of service to the Motor Trade, and to have been associated with so many great names of the industry for so long.



A modern Scammell Constructor 6 x 6 Chassis

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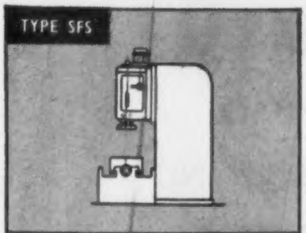
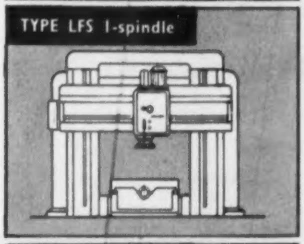
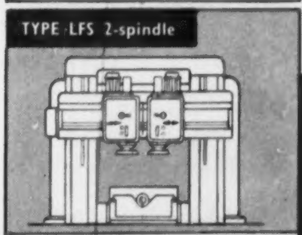
PROPELLER SHAFTS AND UNIVERSAL JOINTS

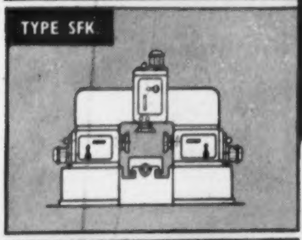
HARDY SPICER LIMITED (*A Birfield Company*), WITTON, BIRMINGHAM, 6
HARDY SPICER (AUSTRALIA) PTY. LIMITED, SOMERS STREET, BURWOOD, E.13, VICTORIA, AUSTRALIA

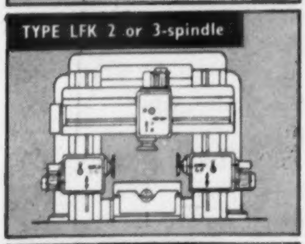
HELLER PLANO-TYPE MILLING MACHINES

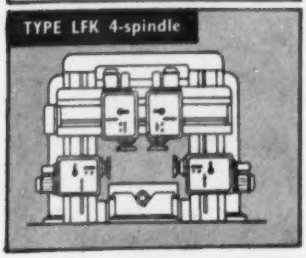




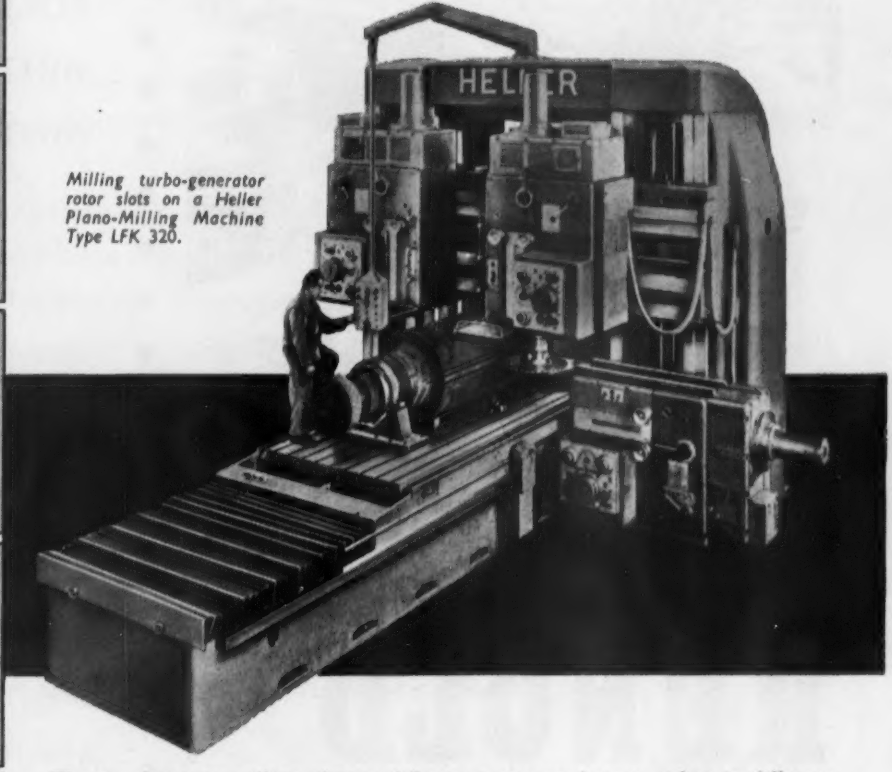








for higher output and greater economy



Milling turbo-generator rotor slots on a Heller Plano-Milling Machine Type LFK 320.

The simultaneous milling of several faces on one workpiece—often in different planes and at different angles—is the answer to the demand for increased output of many complex parts. It obviates re-clamping and re-aligning and in many cases reduces the number of machines required for production.

Heller Plano-type Milling Machines are designed on the unit-assembly system and can be literally 'built around the component'. The major units are the milling heads, tables, feed mechanism, uprights and frames. The selection of units depends entirely on the work required; for example, light milling heads and a large table can be used for bulky workpieces with only small faces to be milled, or alternatively, heavy milling heads can be employed with a small table for heavy milling on small components. The power employed is therefore related to the workpiece and not to machine table size, and milling heads of different sizes can be fitted to the same machine. The economic advantages of this system of construction are therefore extended into actual operating, saving power and floor space.

Some of the most commonly used arrangements are shown opposite. There is a catalogue available which gives full details. Write for a copy now.

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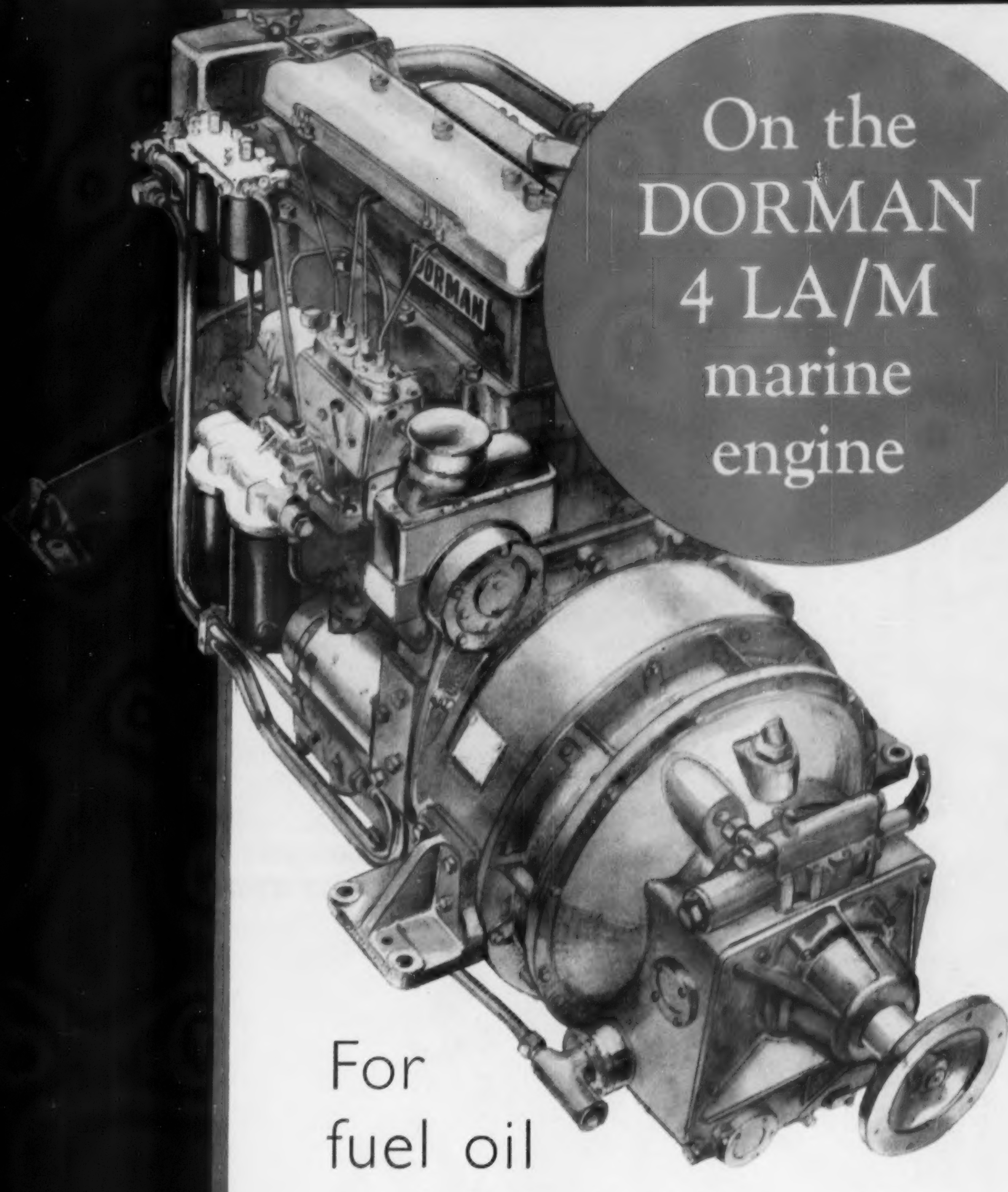

For fuel oil and lubricating oil

This engine is equipped with the Purolator MF.5815 oil filter for the fuel oil and the Purolator MF.4304 filter for the lubricating oil.

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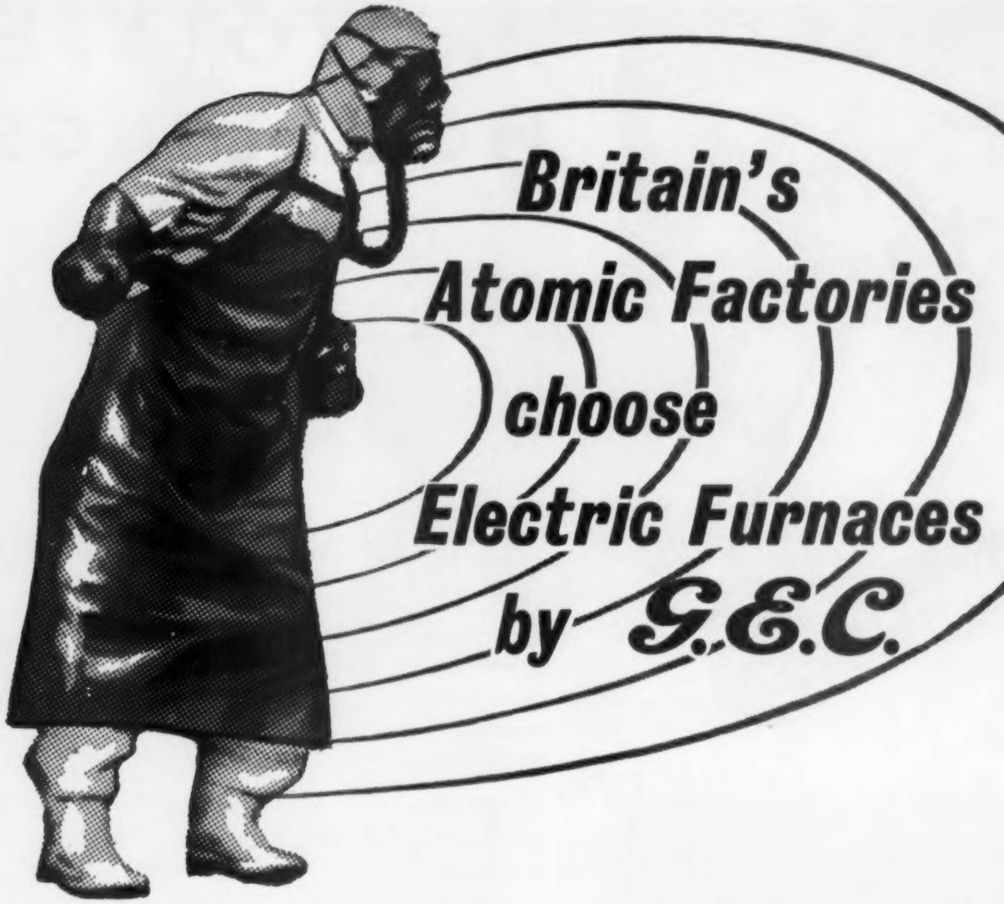
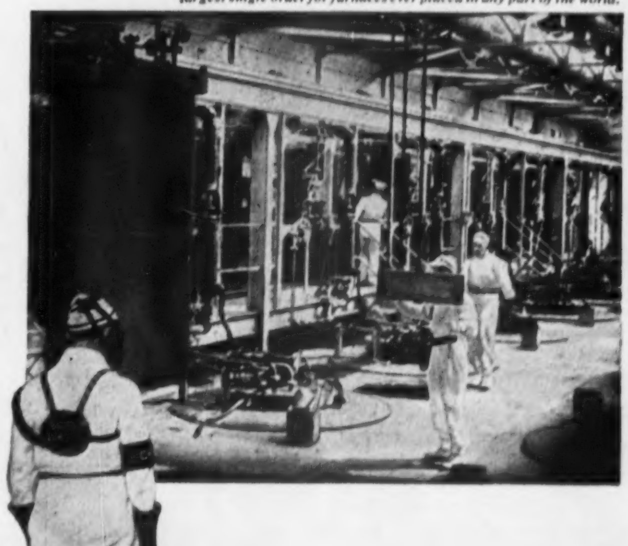


Part of the British Atomic Energy furnace installation. The 54 furnaces and 200 furnace pots comprises what is probably the largest single order for furnaces ever placed in any part of the world.



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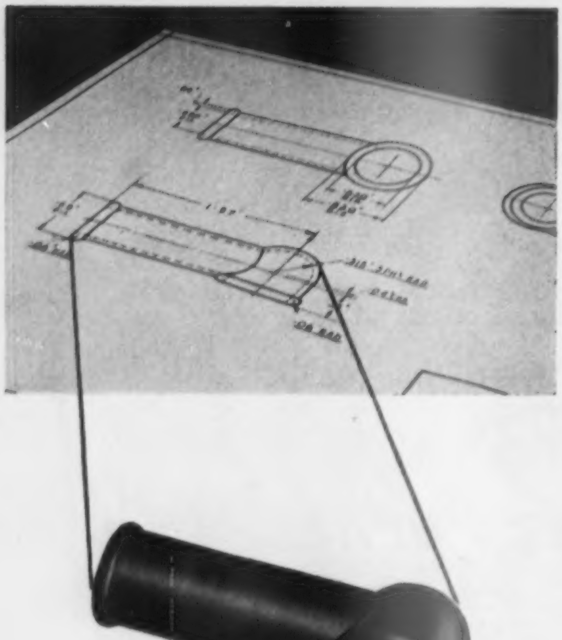

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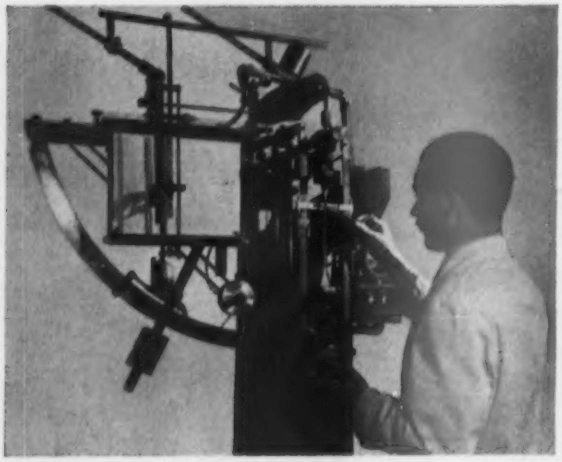

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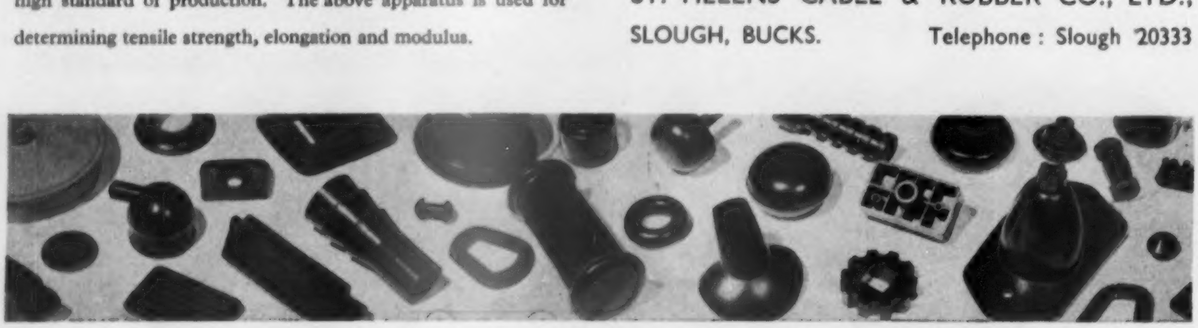

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PEARLITIC	35–45	25–35	1–5	55–65	65–80	4	230/280
PEARLITIC/FERRITIC	32 min.	24 min.	5 min.				
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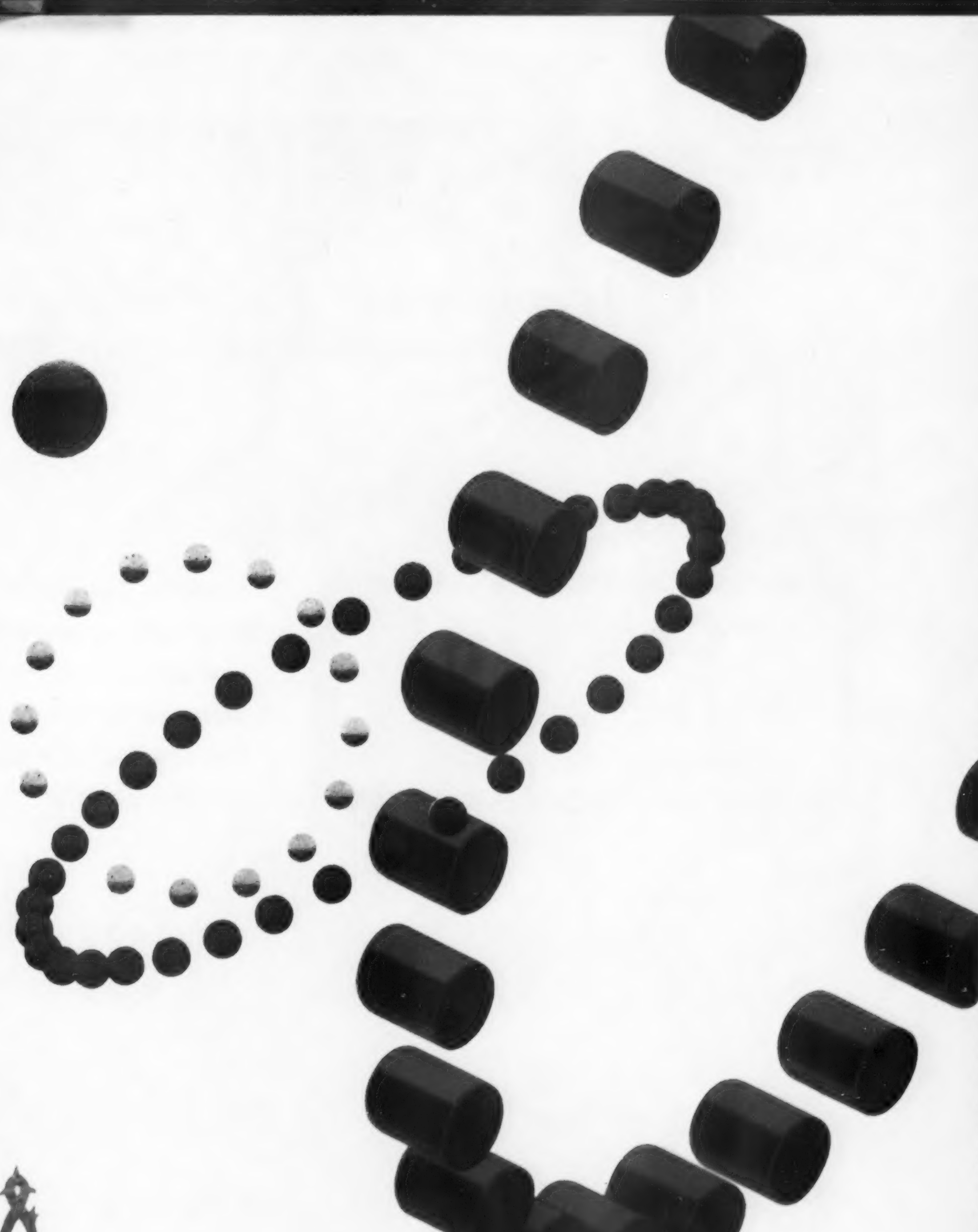

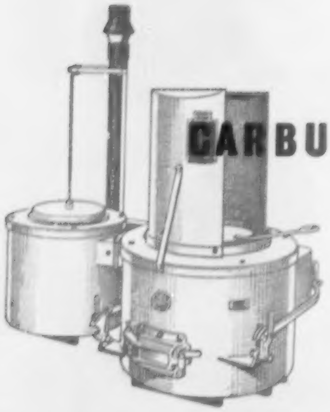

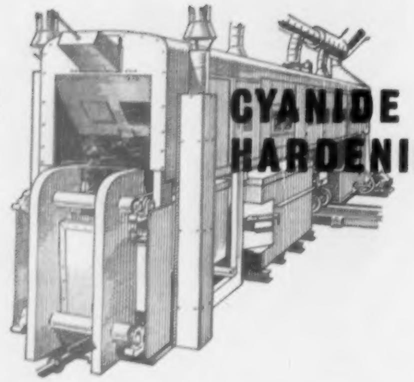

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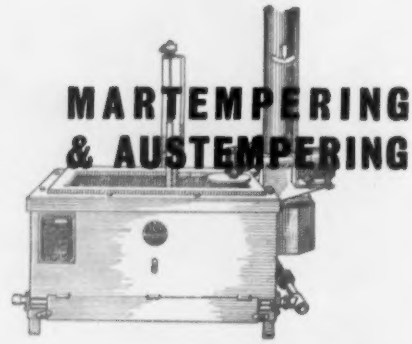

The 'Cassel' Heat-treatment Service

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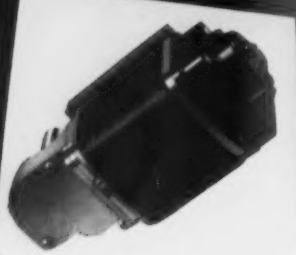

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Windscreen Wipers 1955

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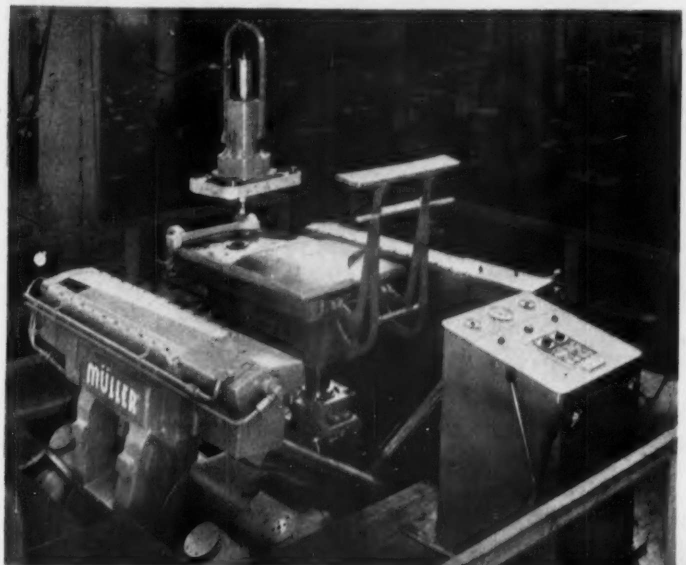

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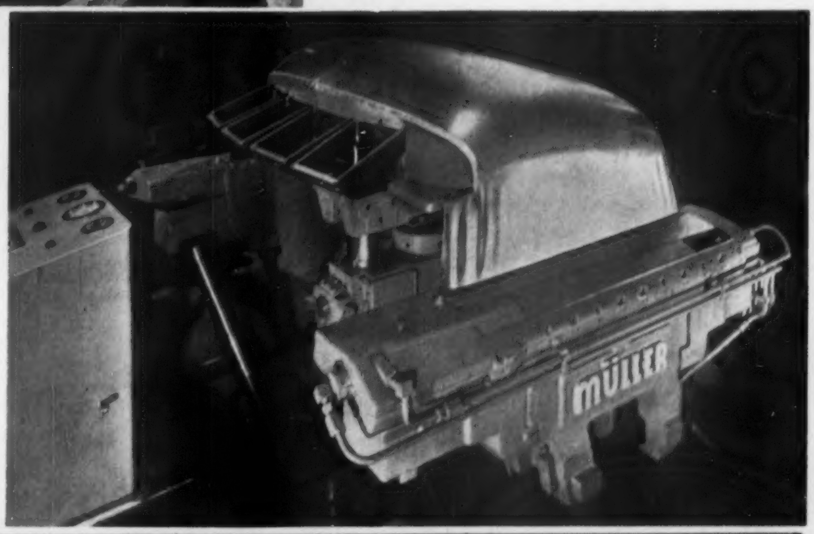

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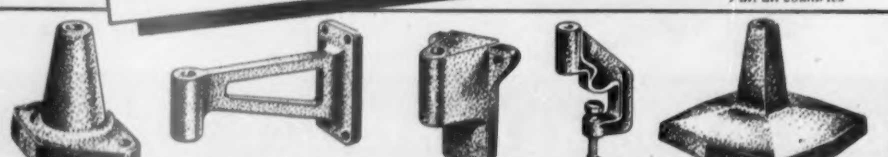

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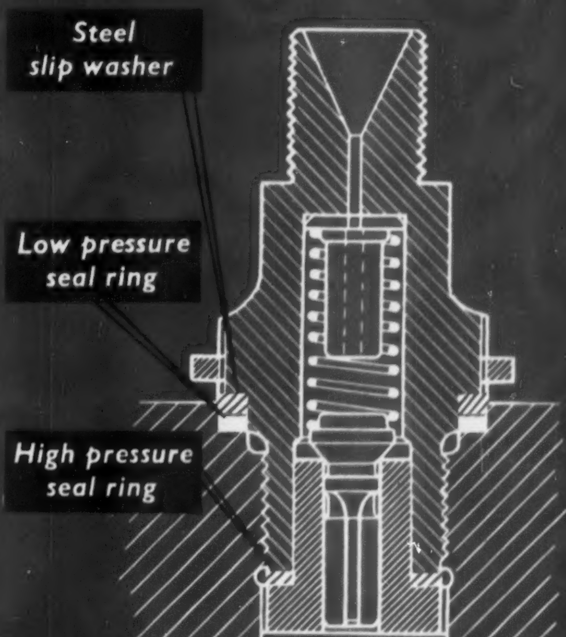

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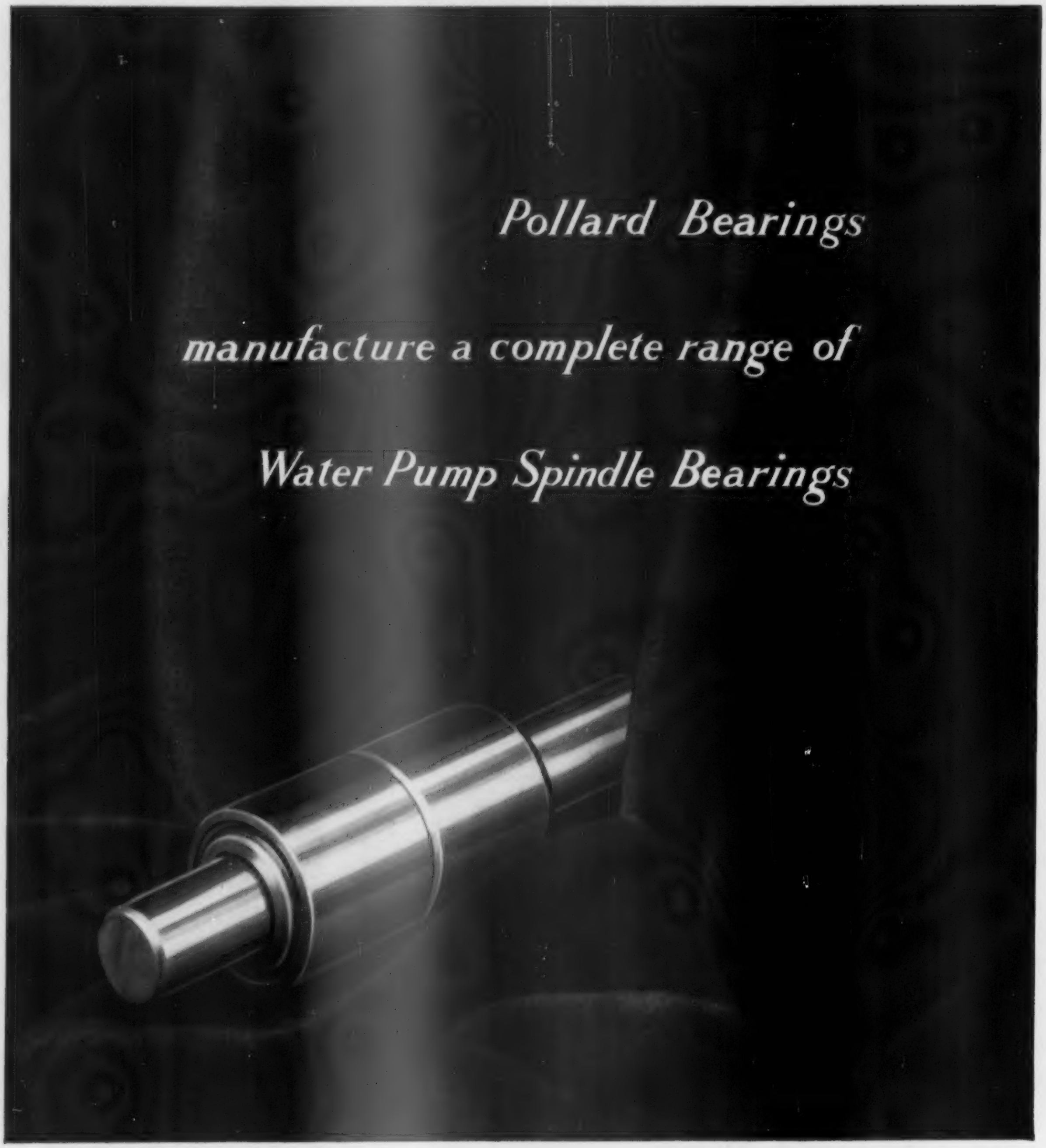
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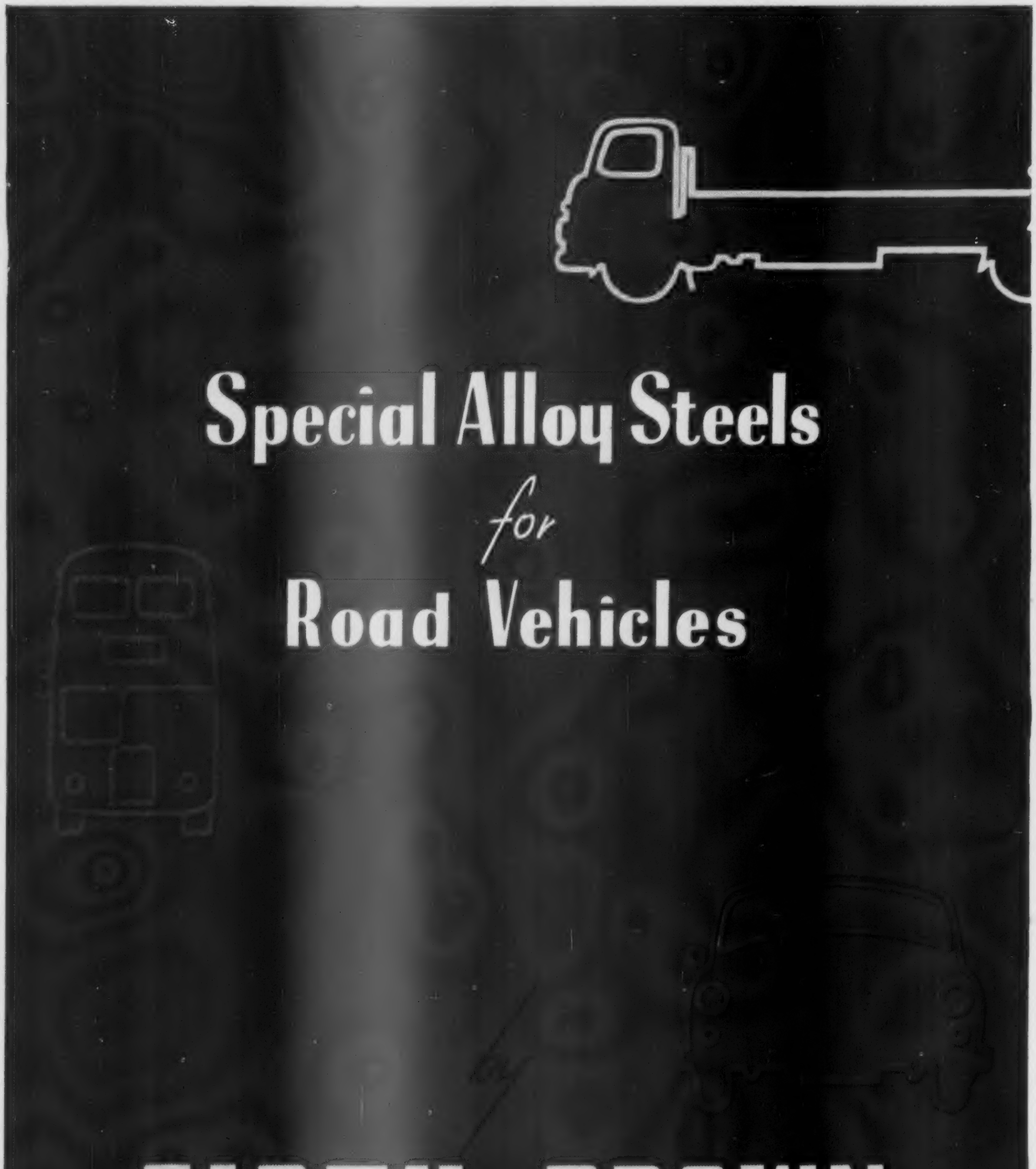
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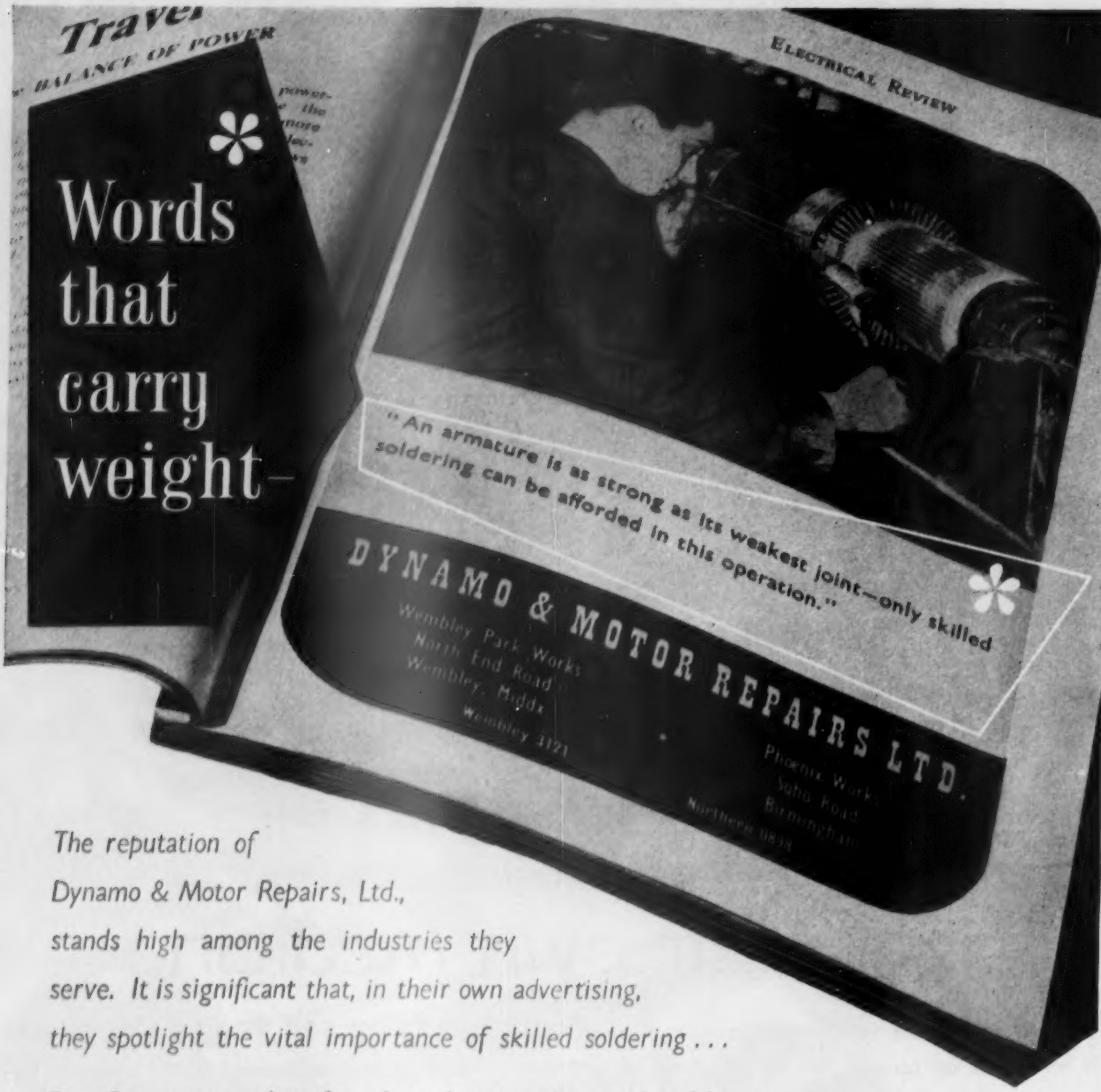

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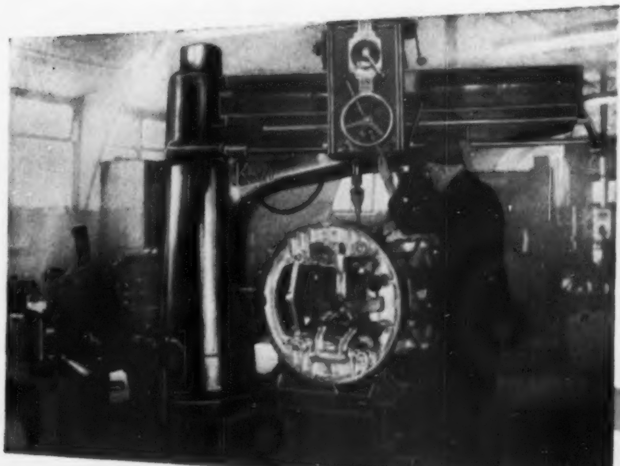

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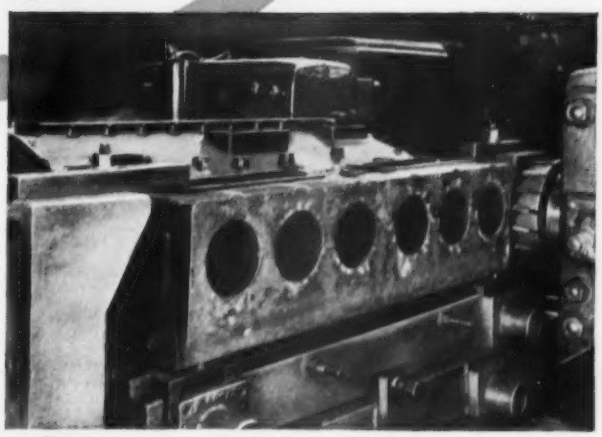

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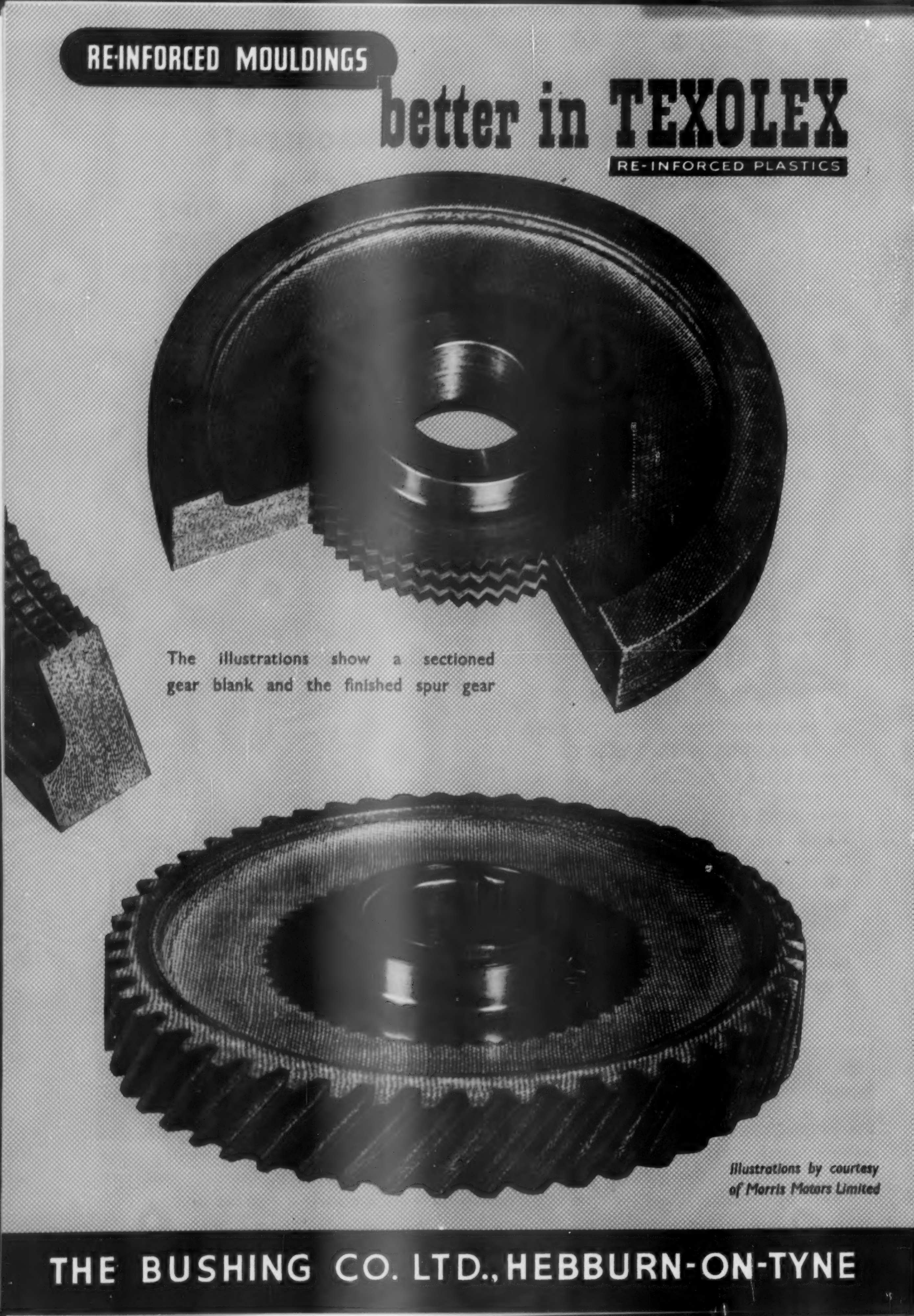

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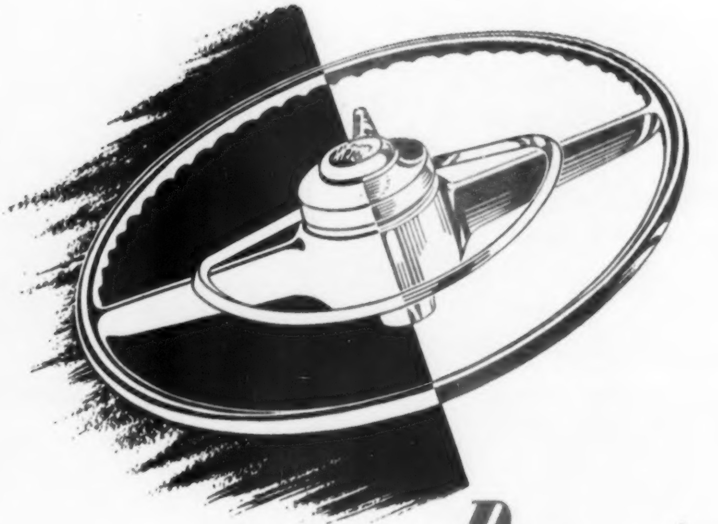


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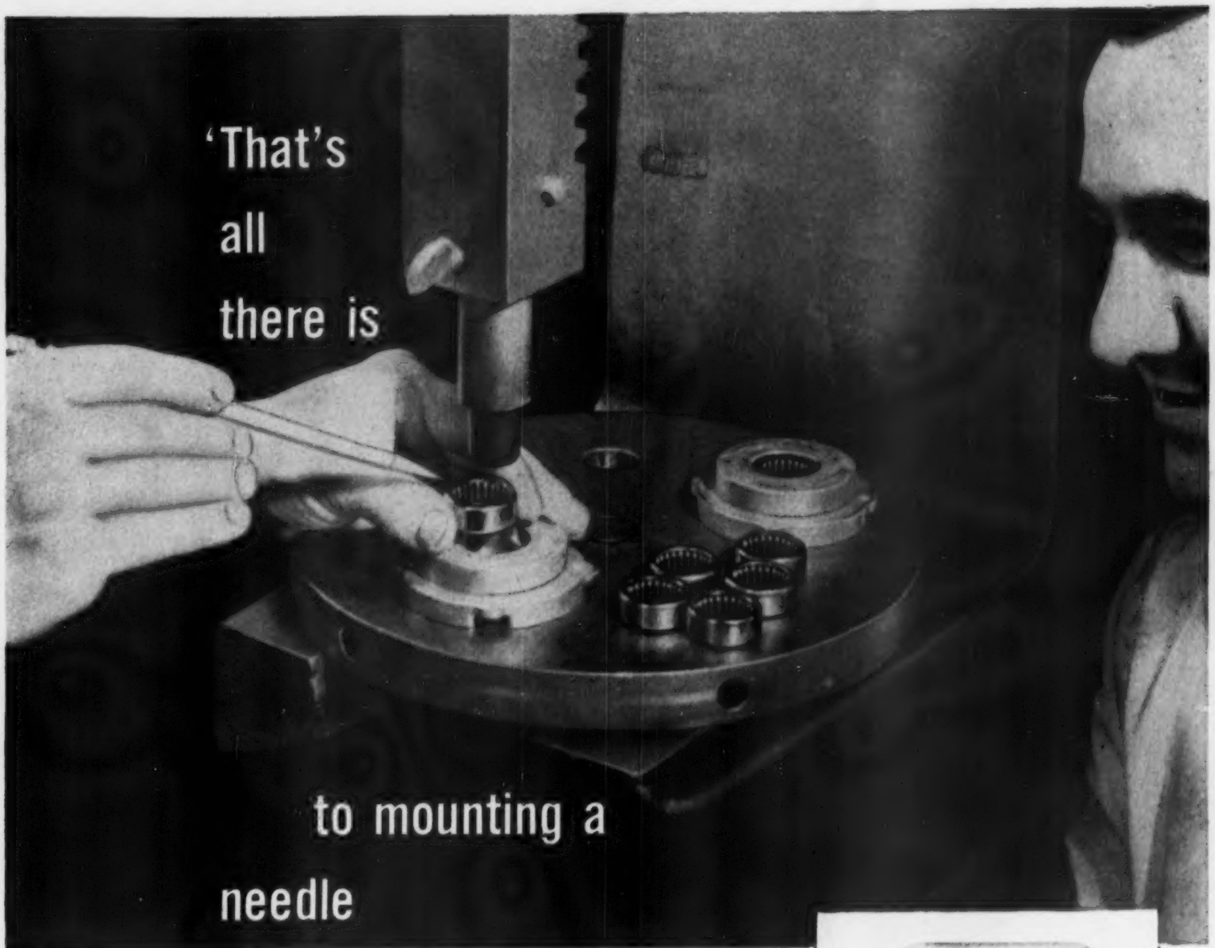

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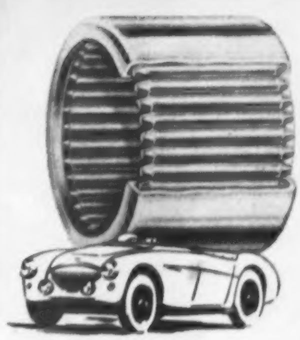

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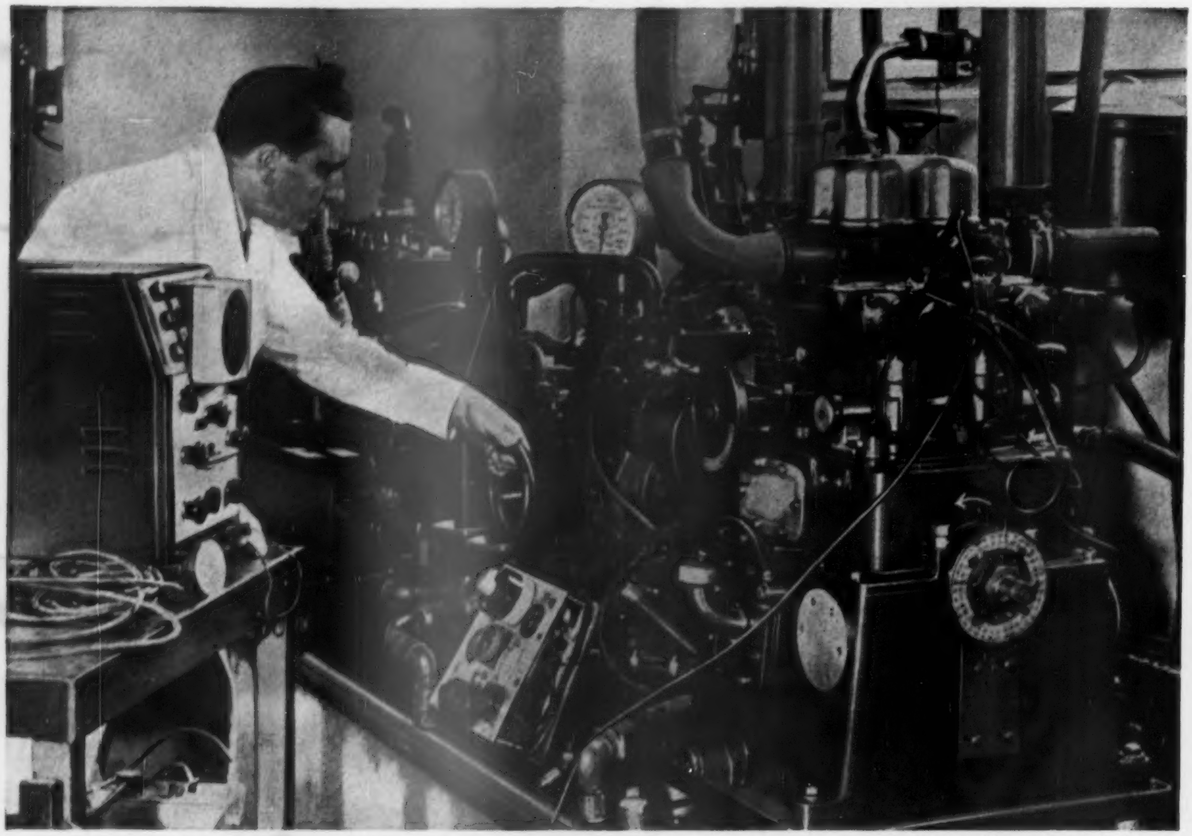

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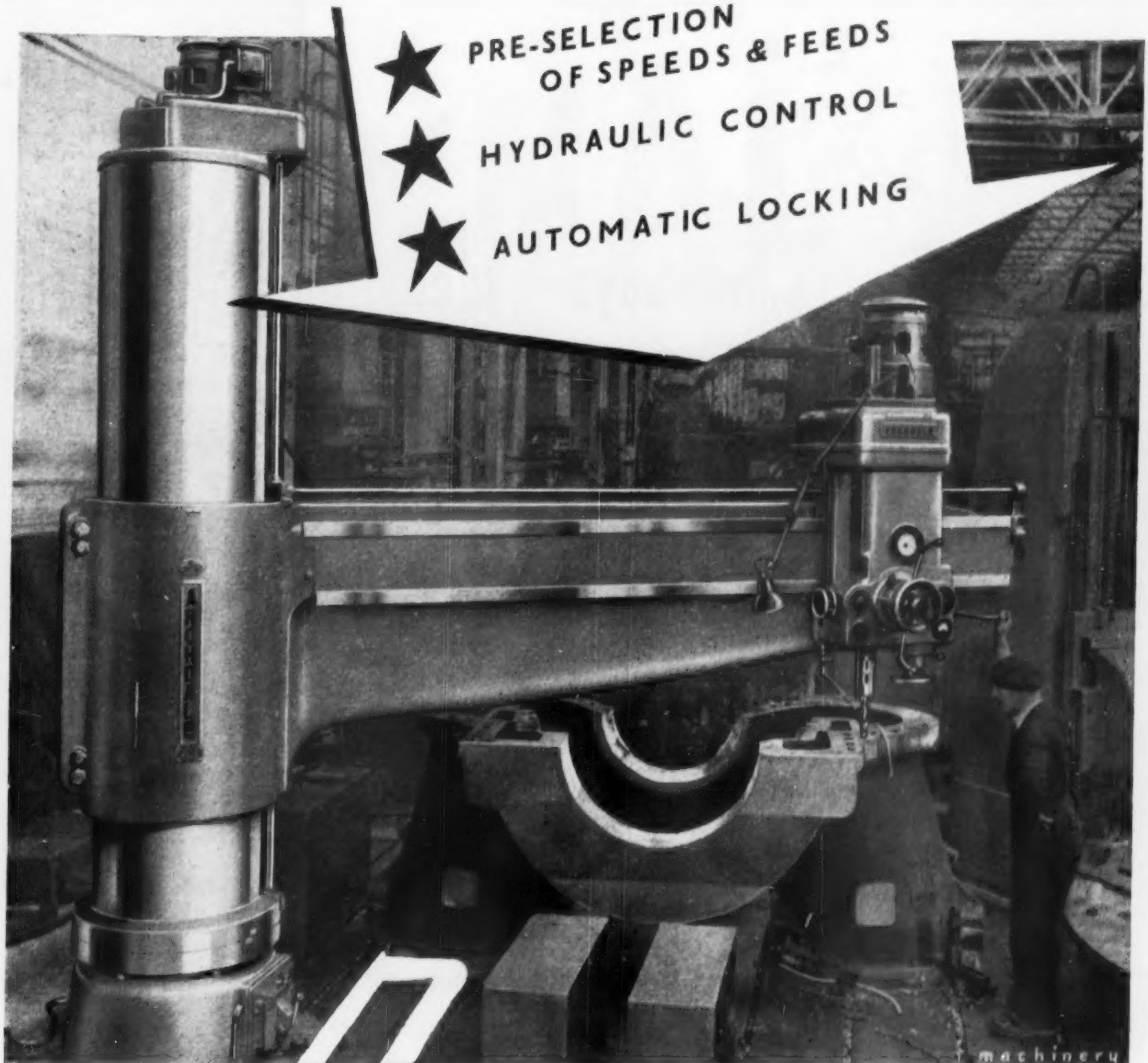

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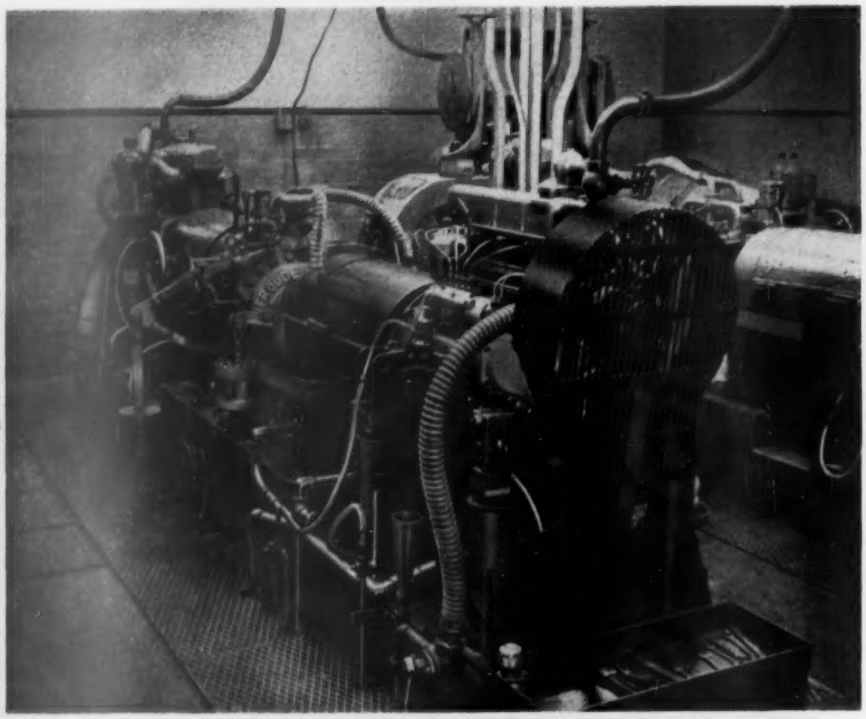

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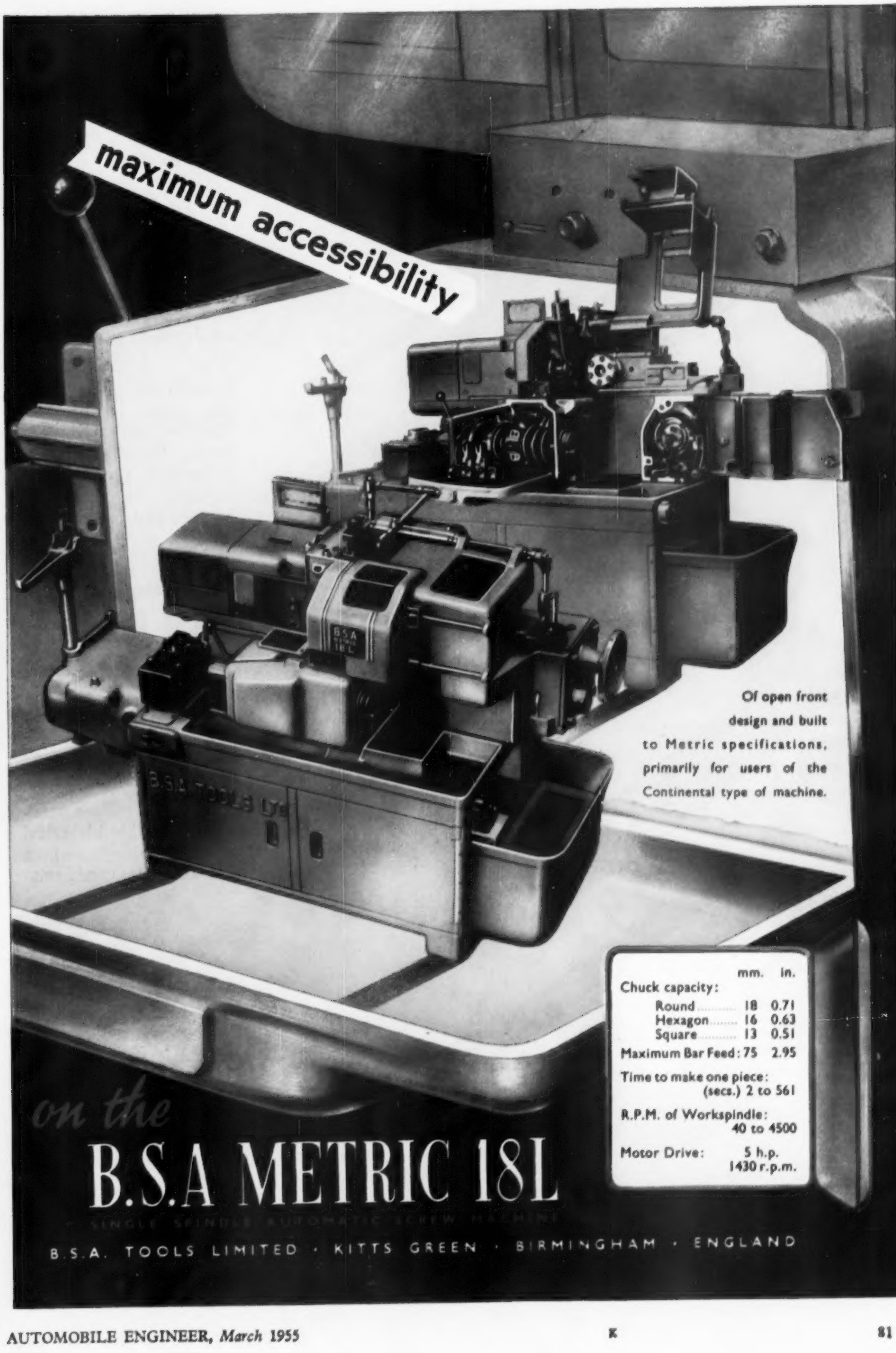
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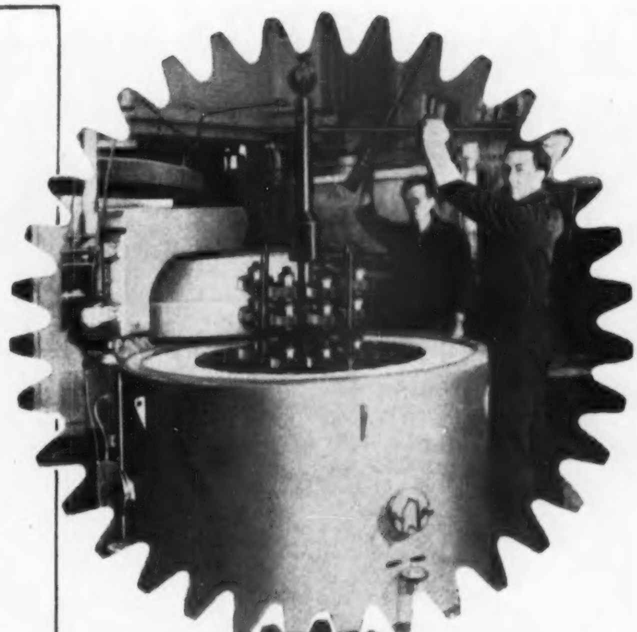

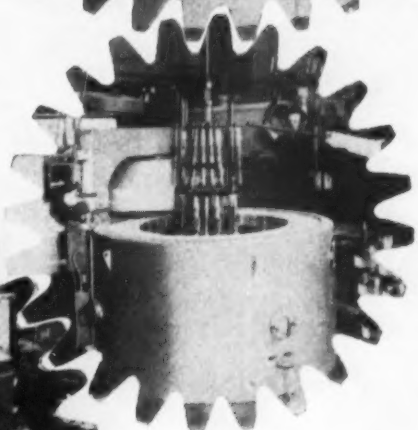

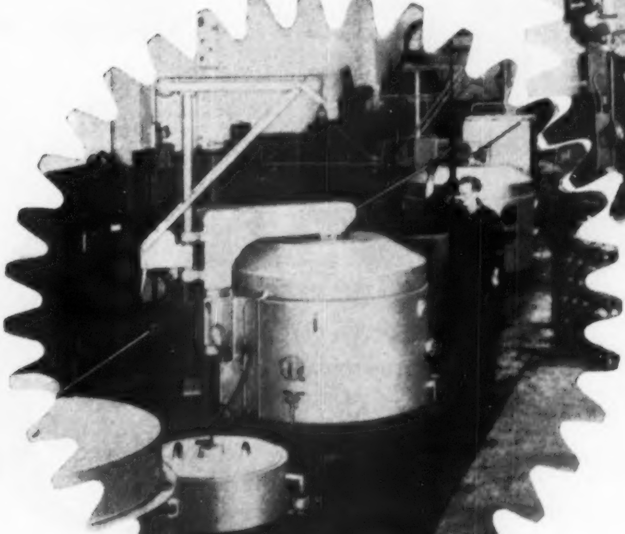

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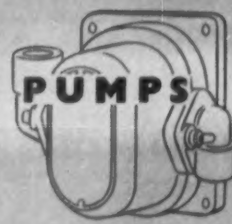

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Design, Materials, Production Methods, and Works Equipment

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Personnel Recruitment

WHEREAS in the past an industrial organization relied almost wholly upon recruitment from the locality in which it was situated, to-day there is evidence that recruits are being sought from farther afield. For example, in recent weeks there has been a series of advertisements in *The Manchester Guardian* dealing with the possibilities open in various large and medium scale industrial organizations. Among others, the advertisers include British Motor Corporation, Joseph Lucas Ltd., British Oxygen Co. Ltd., The Avon Rubber Co. Ltd., and General Electric Co. Ltd. One point that emerges from all the advertisements is an obvious desire to recruit boys and young men with higher scholastic attainments than their pre-war counterparts.

That all the firms taking part in the campaign have adequate training schemes for recruits of all classes is clear. In fact, any boy leaving school or young man leaving university can feel assured of being afforded every assistance towards making the fullest possible use of any talent he may possess. This is all to the good. Every entrant can be sure of acceptance into a grade for which his scholastic training fits him, and yet even those who enter at the lowest training grade have the assurance that they will be given the opportunity for promotion to higher grades if they show the necessary ability and application.

It is not, however, in its effect on recruitment for the shop floor that this campaign is important. What is vastly more important is to attract the boys who have continued their education to obtain the General Certificate of Education at advanced level and young university graduates. The immediate financial rewards offered to both types are adequate; they may, in fact, be described as generous, since the recipients are trainees who are economic liabilities, not assets.

Some conception of the importance that is now given to this subject may be gathered from a brief résumé of the schemes operated by Joseph Lucas Ltd. Trade, engineering and laboratory apprenticeships are available for those who wish to become tradesmen and technicians. These apprenticeships last for five years and start in the Lucas engineering schools.

Public and grammar school boys, with a General Certificate of Education at advanced level are offered, on the completion of National Service, a General Administration Course that extends to two or three years. During the third year there is specialization.

There are post-graduate courses, lasting two years, for arts and science graduates. Arts graduates are given a standard one year course in general administration, followed by a year of specialized training. Engineering graduates take a practical engineering course for the first year, during which they spend five months in the Company's engineering school. In the second year, they cover the administration departments of the organization.

As soon as a trainee's particular interests are revealed, specialist instruction is given. Nor does training end at the completion of the apprenticeship, since supervisory and management training continues at the Company's residential training college associated with the courses at the various factories. In addition, opportunities are provided for promising students to attend external courses in technology and management in this country and overseas.

There is little doubt that continued progress in the automobile and ancillary industries will increasingly depend upon management and technologists. It is, therefore, encouraging that so many organizations are now running schemes that will first attract the right type of recruit and then give the training which will develop the recruit's potentialities.

Commercial Vehicle Suspension

FOR some time, it has been increasingly apparent that conventional suspension systems employed in almost all passenger-carrying commercial vehicles leave much to be desired. However, some manufacturers have shown the way to improvement, notably the Birmingham and Midland Omnibus Co. Ltd., and the London Transport Executive working in conjunction with A.E.C. Ltd., while on the Continent, some vehicles have been fitted at the rear with Gregoire assister-spring type suspensions and anti-roll bars. The primary function of the Gregoire system is to compensate for changes in the load carried by the vehicle. It also helps a little in increasing roll stiffness, but this effect, in most instances, is small and decreases as the load carried is increased. Therefore, if a softer than normal suspension is required, an anti-roll bar is generally an essential adjunct to the system.

The main problem involved in the search for softer suspension is to avoid substantially increased costs in both manufacture and maintenance. It seems that some increase in first cost is unavoidable if better results are to

be obtained. Intelligent design to ensure that servicing operations can be readily performed may well do much to offset the effects of increased complexity on maintenance costs. There are also secondary advantages. Suspension systems giving lower frequency characteristics undoubtedly considerably reduce fatigue loading in the vehicle as a whole. This, together with the adoption of chassisless forms of construction, with which there are no differential or wracking loads between the frame and body, leads to a considerable economy so far as maintenance of the main structure is concerned.

Naturally, in surveying the possibilities for improvement, manufacturers are studying private car suspension systems of all types. While such a study may furnish a valuable background of information concerning the general principles of independent suspension and ride control, it is still essential to give due weight to considerations peculiar to commercial vehicle suspensions. As in private cars, considerable advantages can be gained by the adoption of independent front suspension. In some layouts it gives more room for the engine installation. Furthermore, almost invariably the spring base is greatly increased because, with conventional semi-elliptic leaf spring arrangements, the necessity for providing clearance for the wheels at full lock seriously limits the distance between the springs.

Front suspension

The double transverse wishbone and coil spring type of front suspension that is so widely used on private cars is not necessarily the best for commercial vehicles, although it has advantages in that the whole suspension unit can be mounted on a readily detachable cross member. Nor does it present any difficulty so far as the provision of clearance for the wheels at full lock is concerned. Initially, this type of suspension on cars was developed with metal bushes at the pivots. This was satisfactory until chassisless construction was adopted and the problem of noise reduction arose. Then it became necessary to fit rubber bushes, which incidentally have the advantage of not requiring lubrication. The flexibility of these bushes, coupled with that of the wishbone links, in some instances, has been found to give adverse steering and tyre wear characteristics and it has been necessary to introduce additional links to react brake torque and drag loads.

In commercial vehicles, where loads are so much greater, and particularly in vehicles for export to countries where high average speeds are maintained over rough terrain, it seems likely that a different form of suspension linkage, to give greater stiffness, will be necessary. Stiffness can be obtained, of course, by increasing the cross sectional dimensions of the links; this is not only contrary to the current trend towards weight reduction, but it also increases the unsprung weight, an undesirable feature. It would appear that a better course to adopt would be either to employ a wishbone link system with pivot points spaced wider apart on the vehicle structure, or to arrange the links so that one pair positively reacts the side loads and the other pair the drag loads and brake torque. This is not necessarily the only answer to the problem. Certain engine installation layouts and the relatively great length of commercial vehicles might make practicable in some instances, a leading or trailing link arrangement of a type that is impracticable in private cars.

Rear systems

At the rear also, the main requirement is, of course, to reduce the natural frequency without sacrificing roll stiffness, but the problem is entirely different from that of the front suspension. The final drive has to be accommodated, and is necessarily fairly heavy. Few manufacturers would be prepared at present to go to the expense of fitting a swinging half shaft arrangement if it can possibly be avoided. The layout employed in the London Transport bus, as described in the February 1955 issue of *Automobile Engineer,* is particularly good, but is not the only solution. For instance, several variations of this scheme, with torsion bars instead of coil springs, perhaps of the laminated type, can be visualized. The first impression when studying torsion bar springs, is that their end attachments make them more expensive to install than coil or leaf springs. However, this is not necessarily so with chassisless construction, since the structure necessary to carry these end attachments may be simpler.

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VINCENT INDUSTRIAL ENGINE

An Exceptionally Well Designed 75 cm^3 Two-Stroke Unit

IT is usual to think of industrial engines as being heavy diesel units with bore and stroke dimensions in terms of feet rather than inches. However, there is an ever-increasing demand for small power units suitable for marine applications and for driving pumps, and air compressors for paint-spraying equipment. In the agricultural and horticultural fields, engines of this type are used for cultivators, milking machines and lawn mowers, etc.

To meet this demand, Vincent Engineers (Stevenage) Ltd. are developing a range of small, spark ignition, two-stroke units of 75 cm^3 and 150 cm^3 swept volume. The 75 cm^3 version is in single cylinder form, and the 150 cm^3 units are of twin cylinder layout. The twin cylinder units can be supplied in parallel, V-, or horizontally opposed forms, according to requirements. Nearly all the components are common to the whole range. The manufacturers state that the single cylinder units will be in full production in March, 1955.

Although Vincents are perhaps best known for the manufacture of high quality motor cycles, for many years they have been making components for a wide range of engineering products. As a result, they have gained extensive experience of exceptionally good quality work, such as is demanded, for example, by the aircraft industry, as well as of production on a large scale for the automobile and other industries. It is not surprising, therefore, that there are some novel features in their new industrial engine.

In studying the market for small industrial engines, they came to the conclusion that the designs of most of the two-stroke engines currently available for industrial applications have not kept pace with the rate of engineering progress. In fact, there is little difference between these units and models produced in the early 1920's. Therefore, it seemed that there was room for an entirely new design, aimed at avoiding the well known shortcomings of two-stroke engines in general.

The fundamental requirements that had to be met are as follows: For such a small engine, many of the components have to be manufactured to closer tolerances than for a larger one. Despite this, they must be easy to manufacture, for low cost is even more important in units of this size than in larger ones. Light weight and compactness are also essential, because

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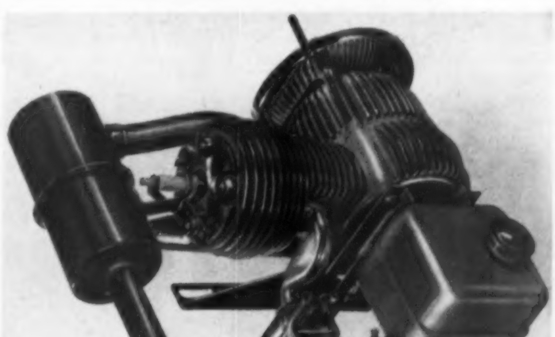
Single cylinder, spark ignition, two-stroke unit. Bore and stroke 43 mm by 50 mm (1·693 in by 1·97 in). Swept volume 75 cm^3 (11·6 in^3). Maximum b.h.p. 1·28 at 2,500 r.p.m. Maximum b.m.e.p. and torque 45·85 lb/in^2 and 3·1 lb-ft at 2,000 r.p.m. Compression ratio 6:1. Air-cooled. Flywheel mounted fan optional. Head, barrel, crankcase, ignition housing all finned. Magneto type ignition. Rotating magnet keyed to crankshaft. Stationary coil. Rotating cam keyed to crankshaft. Contact breaker mechanism mounted on base-plate bolted to laminations. Condenser incorporated in system. Spark plug: K.L.G. F20. Built-up crankshaft. Method of starting: rope on flywheel pulley. Fuel tank capacity: 1 quart. Fuel consumption: 1 pt/b.h.p.-hr. The petrol: oil ratio is 24:1. Carburettor: B.E.C. float type. Dry weight, including fuel tank, silencer and mounting brackets: 24 lb.

portability is a primary requirement for many applications. Another desirable characteristic is easy starting when the engine is hot or cold, and in all climates. This is not a common attribute of two-stroke engines. Since servicing is likely to be done by unskilled personnel, the unit has to be easy to dismantle and assemble, and all nuts and bolts must be readily accessible.

Admittedly, there is difficulty in incorporating all these features in a two-stroke engine, but this type does appear to offer the best prospects of a good compromise between the conflicting requirements. The elimination of valve gear makes a noteworthy contribution to compactness and light weight. It also simplifies the maintenance problem and represents a marked economy, so far as prime cost is concerned, by comparison with engines with valve gear.



A horizontal arrangement of the Vincent industrial engine

Perhaps one of the more serious disadvantages of many two-stroke engines is that they are sometimes difficult to start and do not run smoothly and efficiently under light load. This difficulty arises because, in these circumstances, there is only a small pressure differential between the exhaust and induction systems, so scavenging and charging the cylinder are not effected rapidly enough; consequently, there is a tendency to misfire on alternate strokes, that is, to *four-stroke*. However, these problems evidently are not insurmountable, for the manufacturers state that they have largely overcome them by careful design and development work.

For ease of starting, a rotating magnet type of magneto is employed. This eliminates completely all the pick-up brushes and slip rings associated with rotating coil types. Moreover, the rotating magnet tends to be more robust than a rotating coil assembly and, therefore, is more suitable for operation at high speeds. It is stated that the magneto employed on this engine produces an intense spark at very low r.p.m., which doubtless is partly due to the employment of a magnet of the best quality. This, together with careful mechanical design and good carburation, ensures instant starting under all conditions. The same features give the unit smooth running characteristics and have largely eliminated the tendency to four-stroking.

One of the mechanical design features leading to smooth running and easy starting is the precision machining of the gas ports to maintain accuracy of size, shape and location and to ensure good flow characteristics. This is also an important factor in maintaining the maximum power output throughout the life of the engine. Another feature that leads to smooth running is that all the rotating and reciprocating parts except the interior surface of the die-cast pistons are machined to maintain a high degree of balance. The crankcase volume is reduced to an absolute minimum to ensure that crankcase compression is as high as possible. So far as starting when the engine is hot is concerned, the generous finning round the crankcase helps by keeping that component cool.

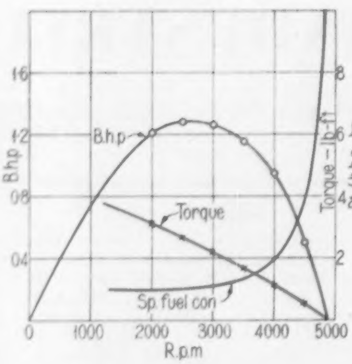
These desirable qualities are examples of the benefits obtained by

a bold policy of accurate machining and using good quality materials and components where they are justified, despite the extra cost involved. There are other examples of this policy: one is the use of a good quality cast iron for the counter-weight, another is the adoption of aluminium for so many of the castings, and a third is the fitting of a really effective silencer.

Manufacturing costs have been reduced to a minimum by careful design for ease of production. For example, the connecting rods are flat steel stampings, which can be ground readily to close tolerances on a turn-table machine. In this way, their weight can be accurately controlled. The counter balance weight, also, is a simple flat component that is easy to produce to accurate limits. A simple mainshaft is employed so that it can be centreless ground. The crankcase and cylinder barrel are in one piece and are made by the shell-moulding method to give a good finish.

An engine has recently completed two 500-hour tests at maximum output. At the end of the first run, the power developed was slightly greater than at the beginning of the test. Doubtless, this was due to a general running in and freeing of tight bearings. Examination of the components after stripping the engine showed that wear had been negligible except on one component.

The defective component was the crank pin. It was unaffected for one-third of the distance from the face of the counter-weight, but the case had flaked over the outer two-thirds of its length. This was attributed to the fact that the core material was not strong enough to support the case, so a new pin of En 36, instead of En 32, was fitted and, in the subsequent 500-hour test on the same engine, neither this



Engine performance curves

component nor any other gave trouble. Part of the development programme now in progress is to get even greater power output from the unit, although that already obtained is good for this type of engine.

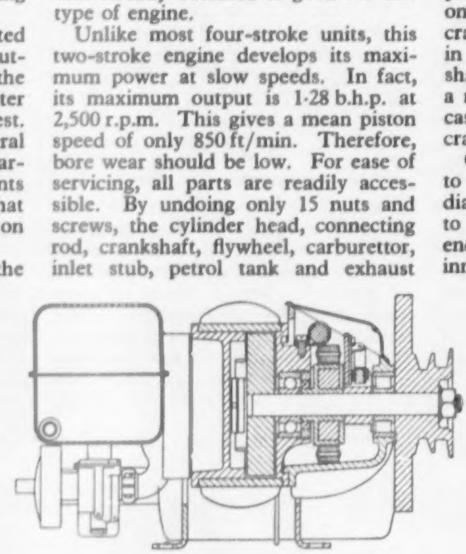
Unlike most four-stroke units, this two-stroke engine develops its maximum power at slow speeds. In fact, its maximum output is 1·28 b.h.p. at 2,500 r.p.m. This gives a mean piston speed of only 850 ft/min. Therefore, bore wear should be low. For ease of servicing, all parts are readily accessible. By undoing only 15 nuts and screws, the cylinder head, connecting rod, crankshaft, flywheel, carburettor, inlet stub, petrol tank and exhaust system can be removed and the magneto dismantled except that the laminations, coil, condenser and contact breaker remain attached to the inner bearing housing.

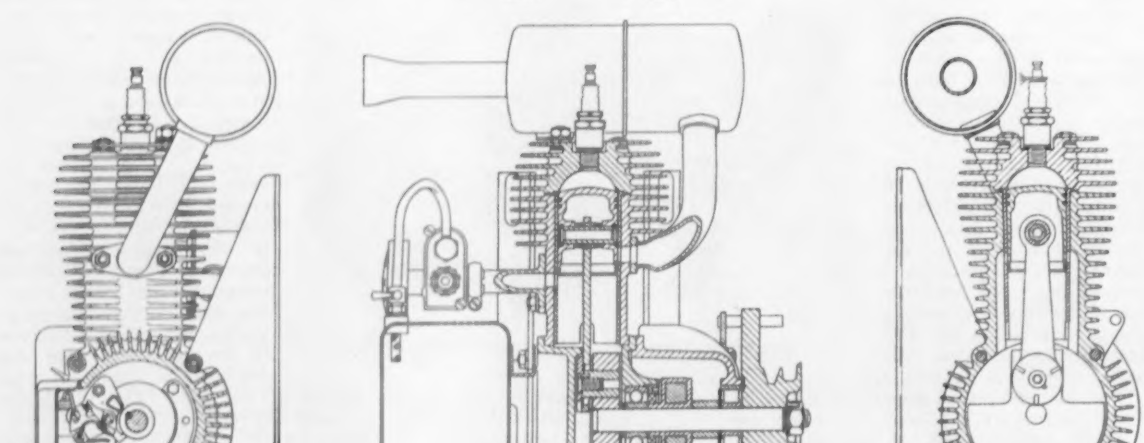
Air cooling has been adopted to cut weight and size to a minimum. Provided the unit is moving, even at the slow speed of a lawn mower, cooling is adequate without a fan. However, for stationary applications, a simple pressed steel fan is mounted on the flywheel. This is adequate without any form of shrouding because of the high thermal coefficient of aluminium alloy that is used for the cylinder head, barrel and crankcase.

General description

The aluminium alloy cylinder barrel and crankcase are cast integrally. This saves machining, simplifies assembly, improves heat flow and makes the unit more rigid than would otherwise be practicable. Two covers are fitted, one on each end of the cylindrical crankcase. These covers are spigoted-in to locate them radially. They are shaped in such a way as to reduce to a minimum the volume in the crankcase and thus to give a high degree of crankcase compression.

One of the covers is of cupped form to project into the crankcase. A large diameter recess is cast in its inner end to clear the path of rotation of the end of the crank pin assembly. The inner face of the other cover is flat, and is just clear of the integral crank web and counter-weight. This cover houses the ball thrust bearing that axially locates the crankshaft assembly. Since the clearance between the covers and the rotating components in the crankcase is only about $\frac{1}{32}$ in, the dimensions between the crankcase joint faces and the inner faces of the covers, as well as the axial location of the shaft





In the Vincent 75 cm³ capacity industrial engine, the components are designed for ease of production to close tolerances and for simplicity of servicing operations

assembly, have to be accurately controlled.

The crank assembly is overhung mounted in two bearings. One is the ball thrust bearing, already mentioned, in the crankcase cover, and the other is a roller bearing in the end of a cup-shaped casting, which encloses the magneto as well as forms the bearing housing. Thus, the magneto is between the two bearings.

This overhung crankshaft arrangement has, of course, been adopted to reduce the crankcase volume to a minimum. It also has the advantage that only one seal is required round the shaft, instead of two, as would be necessary were the crank straddle mounted between two bearings. Moreover, with a fabricated crankshaft, there is less likelihood of relative movement between the components; if the engine is stopped suddenly, for example, by a stone in a lawn mower, misalignment frequently results when there are two crank webs on a built-up shaft. Such a mishap often passes unheeded and the consequent vibration and misalignment cause rapid wear and may lead to loss of goodwill.

The integral crank web and counter-weight is of a high tensile cast iron. It is pressed on to the end of the $\frac{5}{8}$ in diameter En 16T shaft. The crank pin, which is of En 36 case hardening steel, is pressed into the web. An interference fit of 0·0031-0·004 in has been adopted for the shaft and 0·002-0·003 in for the crank pin in the counter-weight.

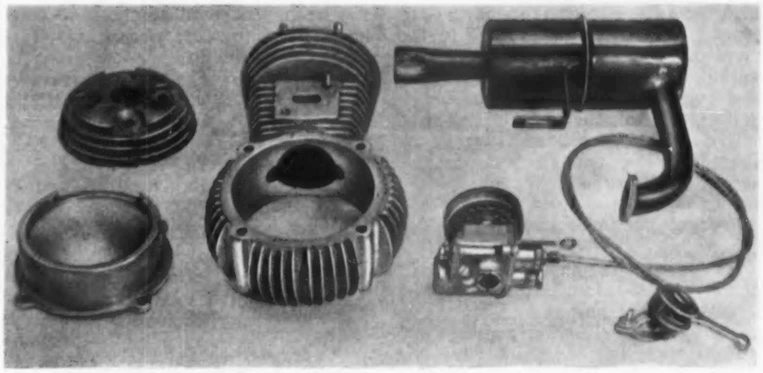
Between the inner race of the ball bearing and the crank web is a distance ring. Outboard of this bearing is the rotating magnet, the contact breaker cam, a thrower disc, a roller bearing and the flywheel. The whole assembly is retained by a nut and tab washer on the $\frac{1}{2}$ in diameter threaded end of the shaft. The auxiliary flywheel is of high tensile cast iron. It is incorporated to give good slow running, but need not be fitted if the engine is used to drive machinery that acts as a flywheel. Axial location of this assembly is effected at the ball thrust bearing, the outer race of which is pulled against a shoulder in its housing by a ring nut. A lip type oil seal is fitted between the inner periphery of this ring nut and a sleeve extension of the magnet housing.

The outer race of the roller bearing is pressed into its housing and is not positively located axially relative to the inner race and roller assembly, which are therefore free to float to allow for thermal expansion. This roller bearing is packed with grease on assembly. Accurate alignment between the two bearings is provided for by a spigot at the joint between the two covers that form their housings.

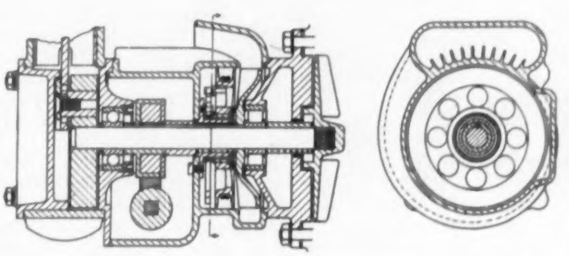
The distance between the bearing centres is approximately $2\frac{3}{4}$ in.

Four $\frac{1}{4}$ in diameter bolts are passed through the crankcase to secure the covers to it. The covers are so designed that they can be fitted to either side of the crankcase to suit the requirements of any particular application. Moreover, they can be assembled in any of four positions by turning them through intervals of 90 deg, so the cylinder can be arranged vertically above the crankcase, horizontally on either side or vertically below it. These modifications can be carried out by the customer provided he has equipment to cut a new keyway in the crankshaft to make the necessary alteration to the magneto timing.

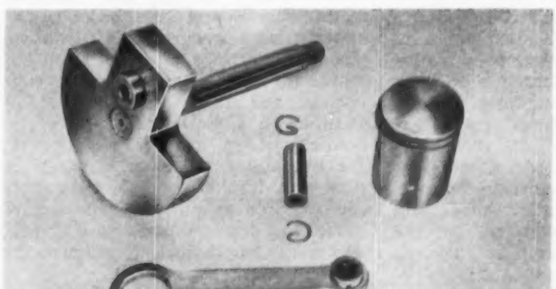
A Ticonal magnet, supplied by Mullard Ltd., is employed. It is



The cylinder block and crankcase assembly, exhaust system and carburettor



As an alternative to the standard layout, this arrangement is offered for mounting the engine directly on the machinery it drives



Crankshaft, connecting rod and piston assembly

Magneto assembly and its casing

enclosed in a zinc die-casting to protect it against rough usage in the event of the unit being dismantled by unskilled hands. It has been found that even after this magnet has been deliberately misused grossly and kept under adverse conditions, it retains its magnetism and still gives satisfactory service in the engine.

Crank pin and connecting rod assembly

The outside diameter of the En 32B crank pin is 0·723 in. A hole is drilled axially in this pin and is tapped to receive a $\frac{3}{16}$ in diameter countersunk-socket-screw that secures the big end retaining plate. This plate is circular, and machined radially in its periphery are three slots spaced 120 deg apart. The slots are incorporated to allow lubricant to enter the bearing. A $\frac{3}{32}$ in diameter peg is driven into a hole drilled through the plate and into the end of the crank pin. This is to prevent relative movement between the plate and the crank pin, which would cause the screw to work loose. The head of the countersunk-screw blanks off the end of the hole and thus prevents the peg from coming out. At the edge of the countersink in the plate, a small slot is milled, and the screw is locked by punching a small portion of the periphery of the head into it.

A simple connecting rod made of $\frac{3}{16}$ in thick, En 32 strip is employed. Fifteen $\frac{3}{16}$ in diameter rollers run directly in the rod and on the pin. Both the pin and the bore of the big end are case hardened. A phosphor bronze bush of substantial proportions is pressed into the small end. Its principal dimensions are: $\frac{11}{16}$ in outside diameter by $\frac{1}{2}$ in inside diameter by 0·575 in long. Two $\frac{3}{32}$ in diameter holes are drilled diametrically through the bush, one each side of the connecting rod, to assist lubrication. The reason why the bush is of relatively thick section is, of course, that its ends are unsupported, and therefore would otherwise tend to bell-mouth.

The piston assembly

The Hepworth and Grandage Kaseit gudgeon pin is $\frac{1}{2}$ in outside diameter × $\frac{3}{32}$ in inside diameter. It is retained by wire circlips in the cross holes in the piston. In each cross hole the bearing length is $\frac{3}{8}$ in. The ends of the pin are machined squarely and are not plugged.

A Hepworth and Grandage 413 aluminium alloy piston with a slightly domed crown is fitted. It carries two square-faced compression rings. Their dimensions are $\frac{3}{32}$ in face width × 0·071–0·065 in radial thickness. The depth of the grooves is 0·09 in and the side clearance of each ring in its groove is 0·003–0·005 in.

The cylinder barrel is shrunk on to the cast iron liner. Positive location of the liner is effected by a $\frac{3}{32}$ in wide × $\frac{3}{32}$ in deep flange at the end adjacent to the cylinder head. The head is spigoted over the flange of the liner and is held down by four $\frac{3}{16}$ in diameter studs. A combustion chamber of hemispherical form is incorporated. The axis of the K.L.G., F20, 14 mm spark plug is in line with that of the cylinder.

Round the cylinder head and barrel, the fins are horizontal and are about $\frac{3}{16}$ in apart. They are, in general, $\frac{13}{16}$ in to 1 in deep by $\frac{3}{32}$ in thick at their roots. Smaller fins are incorporated round the lower part of the cylinder barrel and round the crankcase. A simple pressed steel fan can be fitted if necessary. It blows air round the crankcase and the lower part of the cylinder barrel only, so a considerable proportion of the heat flow from the head passes down the barrel. It is, of course, a mistake to overcool the head, since loss of thermal efficiency results.

Other features

The carburettor is supplied by the Bletchley Engineering Co. Ltd., of Slough, Bucks. A special feature is its cold-starting device. Beside the float chamber is another chamber, and the two are interconnected by a duct, which is normally closed by a spring-loaded needle valve. For cold starting, a simple hand-operated control is pulled for five seconds to raise this needle valve off its seat and allow the starting chamber to fill to the level of the float chamber. When the engine is being started, petrol is drawn from this chamber through a jet that is larger than the main jet. By the time the starting chamber is empty, the engine should be running well enough to take over on the main jet. This carburettor is noteworthy for its extreme simplicity: there are no needle or pilot jets and so there is no danger of trouble due to maladjustment during service.

It has been found that the power output of the unit generally is appreciably more than the 1·28 b.h.p. guaranteed by the manufacturers. In fact, with prototype engines, outputs of 1·4 to 1·6 b.h.p. have been obtained. Further development is under way, and it is confidently expected that even better results will be obtained.

Conversion to the twin cylinder form is effected in a simple manner. A modified housing for the magneto and crankshaft bearings is fitted. The cylinder and crankcase units are bolted one on each side of this housing and can be arranged side by side, horizontally opposed, or in V-form.

AUTOMATIC TRANSMISSIONS

THE subject of automatic transmissions is discussed by K. S. Duncan in an article in the *Inst. Automotive and Aeronautical Engineers Journal*, June 1954. The author begins by enumerating the fundamental requirements of a motor vehicle transmission; that is, controllable torsional coupling or clutch, variable ratio torque converter and a reversing mechanism. He also briefly considers the system which immediately preceded the modern automatic transmission, namely, the pre-selector gearbox used in conjunction with the fluid flywheel or fluid coupling.

As used in automatic transmissions, the torque converter is a hydraulic unit comprising a pump driven by the engine, a turbine driven by oil from the pump, and a stator mounted on a one-way clutch. These three elements are enclosed in a housing completely filled with fluid. The design of each element is described, and an explanation is given of how the fluid circulates from the pump to the turbine and thence through the stator back to the pump. Factors determining the force of fluid flow are considered, and the functioning of the unit is explained by examination of the shape of the blades in each element. A graph shows the characteristics of a combined hydrokinetic torque converter and fluid coupling.

Planetary gear sets are suitable for use in automatic transmissions because the gears are in constant mesh, and ratio changes are effected by the operation of slipping clutches or braking bands on drums. Hydraulic actuating units may be used to make smooth, noiseless changes. Diagrams show the compound planetary gear unit employed in the Ford automatic transmission and the controls for such a unit.

Control systems are based on positive displacement hydraulic circuits. Their operation is explained as follows: "Signals" are received, by the control circuit, from manual pilot control, engine speed, drive shaft speed, throttle position and induction manifold pressure; these signals are automatically integrated to produce actuations at the control points to secure the correct ratio of torque conversion and the desired degree of coupling in any particular set of circumstances. The basic elements in the hydraulic circuit are a pump or pumps to develop pressure and supply the required quantity of oil, control valves to direct the pressure and flow, and hydraulic motors to produce the actuations.

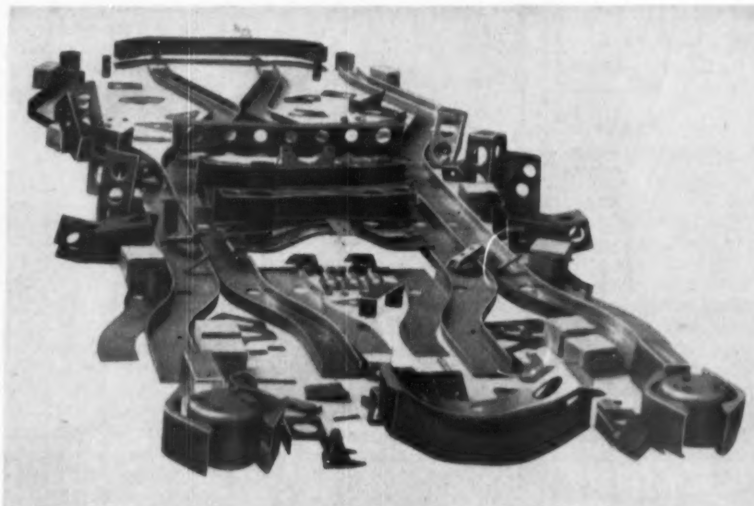
Current examples of automatic transmissions are described; the Hydra-Matic, as used in Cadillac cars, the Ford Mercury Automatic Transmission, and the Chevrolet Power-Glide. Each description is accompanied by a rough classification of the system according to the mechanical and hydraulic elements that are employed. *M.I.R.A. Abstract No.* 6936.

CHASSIS FRAME MANUFACTURE

Production Methods at John Thompson Motor Pressings Ltd.
Part I. Passenger Car Frames

JUST over fifty years ago, John Thompson Ltd., of Wolverhampton, first built motor car frames in one corner of its boiler works for firms no longer in existence. An early customer was the Rover Company, one of the few firms requiring frames in any quantity prior to the First World War and this association has been maintained to the present day. During this period car frames have evolved from simple structures of straight channel-section side and cross members riveted together, into relatively complex constructions of elaborately shaped members welded up to form a single composite unit. Modern specifications stipulate lightness and rigidity in a frame, requirements that are met by reducing the gauge of the material and forming most or all structural members into some type of boxed section to impart the necessary stiffness.

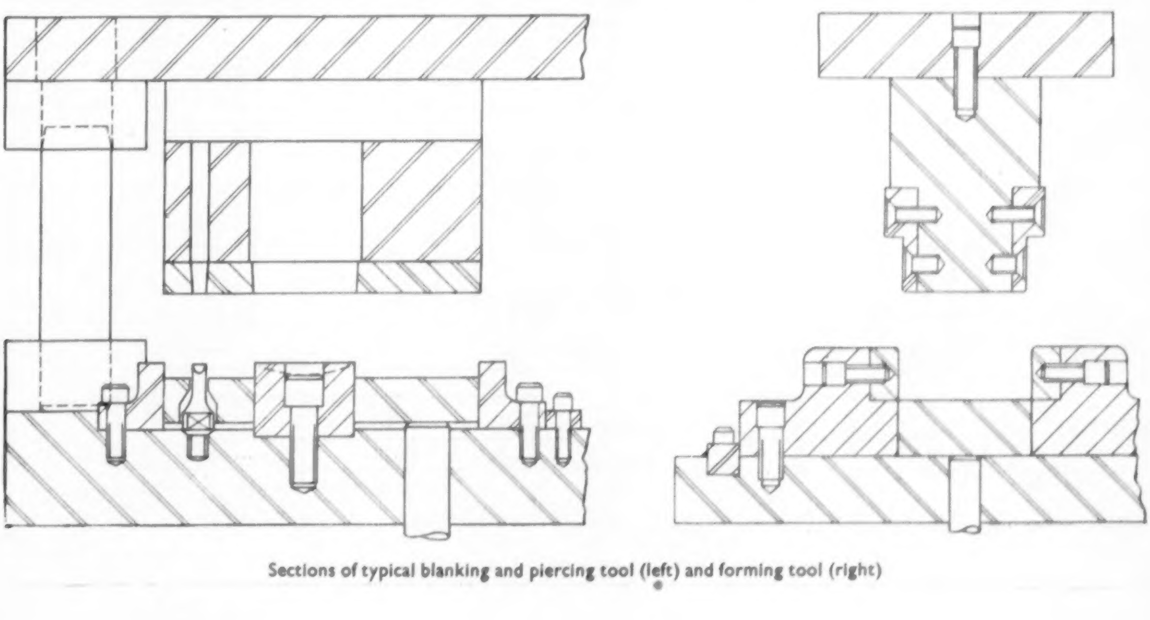
Commonly, a frame will consist of from 100 to 200 component parts, each of which will require individual press tools in some form or another for manufacture. The current Rover frame, for example, is built up of 176 parts and weighs only approximately 220 lb.



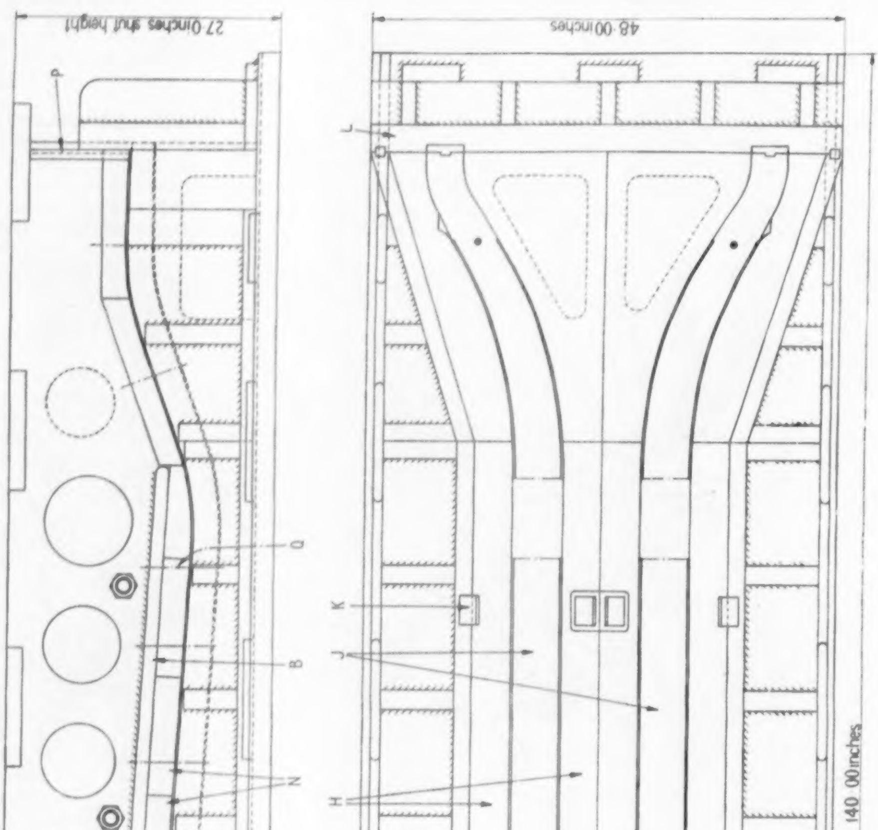
Component parts of a Rover frame

Generally the side members, the most important structural elements of the frame, are upswept at both front and rear ends to accommodate the suspension. The advent of independent front wheel suspension, displacing the hitherto conventional front axle beam, has led to more elaborate front cross members to impart the necessary rigidity to the front end of the frame. Cross members at other positions have also become more complex and must be designed specifically to suit the chassis layout.

A major change has occurred in the method of assembly. In the past the relatively few component members were held together by a few slave bolts and then riveted up. Today each frame requires a specially designed assembly jig to accept and position accurately the various members, sub - assemblies and minor parts which are invariably welded into a composite whole. Under modern conditions, the technique of frame making has become highly specialized. In order that the frame is acceptable on the chassis assembly line, the manufacturer imposes close dimensional and positional tolerances. These necessitate an



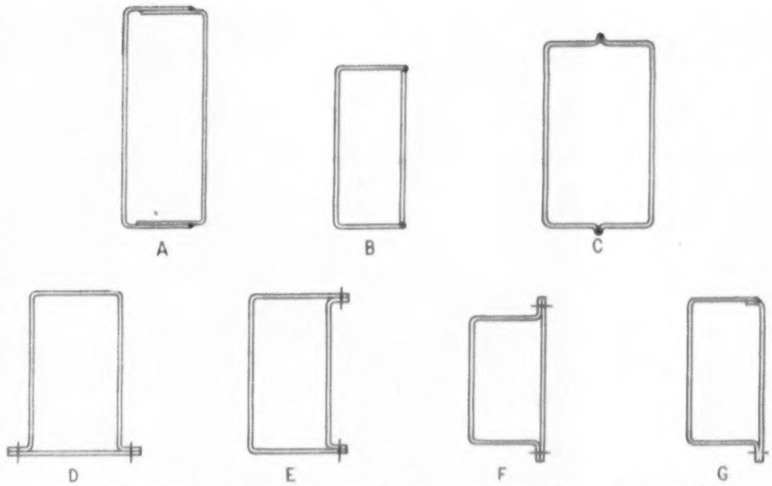
Sections of typical blanking and piercing tool (left) and forming tool (right)



TYPICAL SIDE MEMBER FORM DIES

A—Base. B—Keys. C—Ribs. D—Location pins. E—Tie rods. F—Spacer tubes. G—Plates. H—Die blocks. J—Ejector plates. K—Location blocks. L—End blocks. M—Punch. N—Steelings. P—Punch slide

intimate initial co-operation between customer and producer, special tooling, and meticulous care in assembly and welding. Production currently includes frames and components for Alvis, Austin-Healey, Daimler, M.G., Riley, Rover and Land Rover, Wolseley, and other well-known cars.



A and B—Arc weld. C—Carbon arc weld. D, E and F—Spot weld. G—Arc and spot welds

Light-gauge fabricated box sections

Toolmaking

All tools are made at the plant in an exceptionally large toolroom. For blank and pierce and form tools, mild steel is widely used for the larger sections, with tool steel inserts where necessary.

The mild steel sections are flame cut to approximate shape from slabs or billets by B.O.C. equipment and are passed to planing machines for the first operations of establishing the working datum faces. They are then marked out for profile and passed to shaping or milling machines for the next operation. This is followed by the fitting of the edge "steeling" and finally the various sections are assembled on their respective bolsters.

The toolroom is well equipped with high class machine tools to enable the company to maintain the high standard of die-making required by the automobile industry. Included in the machines is a 6 ft × 6 ft × 14 ft Kendall and Gent plano-milling machine equipped with vertical and horizontal milling head and also a circular table. A Cincinnati Hydro-Tel milling machine with work capacity 120 in × 28 in is used for die sinking and profiling. The latest type 20 in stroke Butler shaping machines with traversing heads are also used for profiling. A Lindner jig borer capable of working to limits of 0·0005 in, with optical equipment for co-ordinate measurement, is housed in a temperature-controlled room having access doors on both sides to admit work of exceptional lengths.

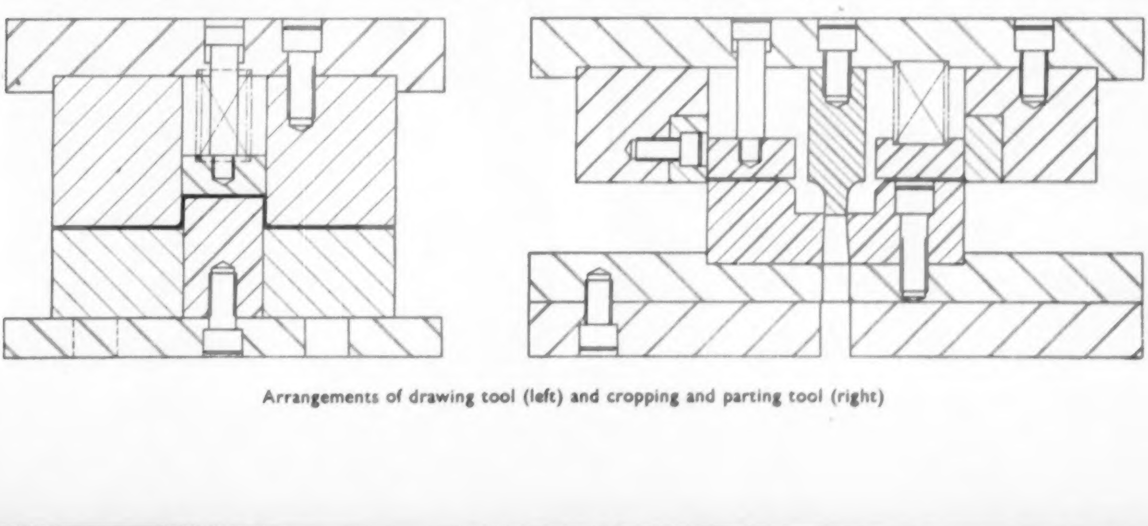
Presswork

With the exception of a few machined parts all component items are produced on presses which range from 25- to 2,000-ton capacity. The majority of parts are produced in two operations, blanking and forming, but certain items may require a third operation to clip the flanges. Further operations, however, may be required on some components according to the design. Wherever possible, holes are pierced when blanking or clipping, or at both operations, but in some instances this is not practical as their position may be affected by the stretching of the material whilst forming. In such cases the holes are later pierced separately or jig-drilled. The larger holes pierced in the blank can be used to obtain positive location of the part on the form tool. Where piercing is done whilst blanking, the punches are mounted in the blanking die and the dies in the blanking punch, as shown in the drawing of a typical tool. Additionally, shear blades may be incorporated to cut the scrap metal from the blank into reasonable lengths for ease of disposal.

All car frame parts are cold blanked and formed on mechanical presses. The largest of these is a specially designed British Clearing press, exerting a pressure of 2,000 tons at 3½ in above the bottom of its stroke; the point of forming. It has a total weight of 400 tons and the 8½-ton flywheel is driven by a 150 h.p. motor. The pneumatic cushion, providing 500 tons pad pressure, is furnished with hydraulic locks. This press can accept work up to 29 ft 6 in long.

Two other presses are normally rated to exert 2,000 tons pressure at the bottom of the stroke. With capacity to accept work up to 25 ft and 20 ft in length they are driven by 120 h.p. and 100 h.p. motors respectively. Hydro-pneumatic cushions each give a pad pressure of 400 tons. Typical of the smaller presses are 750-ton Clearing and 800-ton Wilkins and Mitchell presses accommodating respectively work up to 8 ft and 6 ft in length. Small components are produced on mechanical presses having capacities ranging from 25 to 200 tons.

Great pressure is required to blank out a complete side member as the blank perimeter may be as much as 35 ft in length. The perimeter length is an important factor, in addition to those of material specification and thickness, when selecting a press to perform specific work. Conversely, a particular press can blank or form a straight side frame of greater length than that of one with a kick-up. Where the blank is so large, or more than one blank is cut simultaneously, and the pressure needed approaches the rated capacity of the press, it is the practice to serrate the cutting edges of the blanking die "steelings" and give "shear" effect to reduce the pressure required. The angle of shear is


Arrangements of drawing tool (left) and cropping and parting tool (right)

determined by the material thickness and specification and may range from approximately 2 deg to 6 deg; the larger angles being used for relatively thicker material of higher grade.

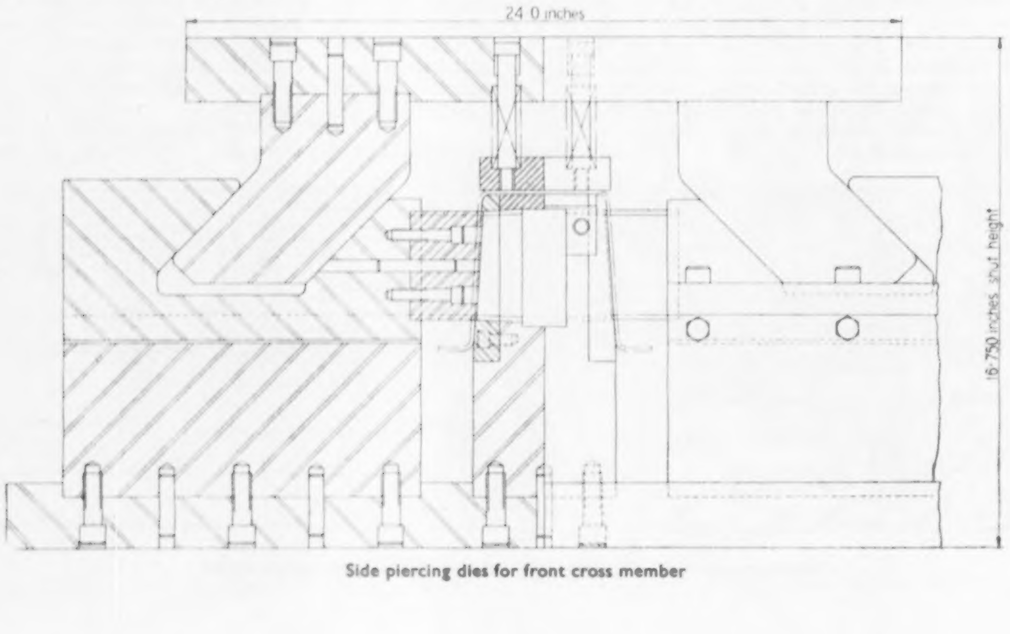
Full-length punches and die sidepieces are sometimes used but more commonly the tools are built in split sections and securely locked to full-length mild steel bolsters. Thus handling, maintenance and storage of tools is facilitated and alternative centre sections can be fitted to produce side members for vehicles of different wheelbase. The locking strips, preventing lateral displacement of the sidepieces, are rebated and bolted or welded to the bolster. Guide pillars and housings are fitted to punch and die sections to ensure correct register when the press is in operation. Punches are invariably made of mild steel and sidepieces of cast iron. The working edges and surfaces of both sections are fitted with "steelings" of tool steel to withstand the high pressure and abrasive wear in use. Ejector plates are of mild steel. The first step in laying out the tools is to determine the position of the die in relation to the ejection plungers in the press table. This is necessary to ensure an adequate, distributed pressure to eject the work from the tool and to obviate the danger of distorting the work by applying a heavy, localized pressure when ejecting.

Specially rated 2,000-ton British Clearing press

Where large quantities of a particular pressing are required, the practice is to use a double tool for forming or clipping. As convenient to the size of the work and of the press bed, they are arranged either side-by-side or end-to-end and a right-hand and a left-hand member are produced at each stroke of the press. For smaller quantities, it is usual to employ a less costly tooling method. A reversible single punch and die is used to press a batch of right-hand members and then the tools are turned over to press a similar batch of left-hand members.

With the object of minimizing any loss of production in event of a press becoming temporarily unusable, ejection holes in all tools are, as far as is possible, arranged to a common pattern so that tools may be transferred to another press. Continuous flow production is achieved for very long runs by using one press for blanking and piercing, a second for forming and a third for clipping.

For certain smaller components, say a front suspension member, requiring multiple operations to complete, several



Side piercing dies for front cross member

sets of tools may be assembled on one of the larger capacity presses and several sequential operations performed at each stroke. Such members are required to be exceptionally rigid and dimensionally accurate and usually comprise a deep pressing with solid drawn ends to receive the helical spring mounting assembly. The material used is pickled and oiled mild steel sheet of extra deep-drawing quality. Its tensile strength will probably not exceed about 18 ton/in^2 as the prime requisites are the essential ductility and surface perfection. As an example, an illustration shows six stages in operational sequence in the production of an M.G. front suspension member. The plate is first blanked and annealed, then follow four pressing operations and a final crop, pierce and side-pierce operation. Simple cross members are produced on the smaller capacity presses by methods similar to those employed for side members up to the sub-assembly stage.

Double form tools for side members

The materials commonly used for car frame components are pickled and oiled general-purpose mild steel sheets, of En 2A, En 2B and En 2C grades, in thicknesses ranging from 16 S.W.G. to 12 S.W.G. It is received from the steel-makers in bundles of sheets cut to dimensions that are multiples of the blank sizes of one or more components. All steel, bar and slab as well as sheet, is held in a huge enclosed store served by overhead cranes. To obviate the possibility of loss of production in the event of a temporary interruption of supplies, a fluctuating sheet stock of a considerable tonnage is maintained.

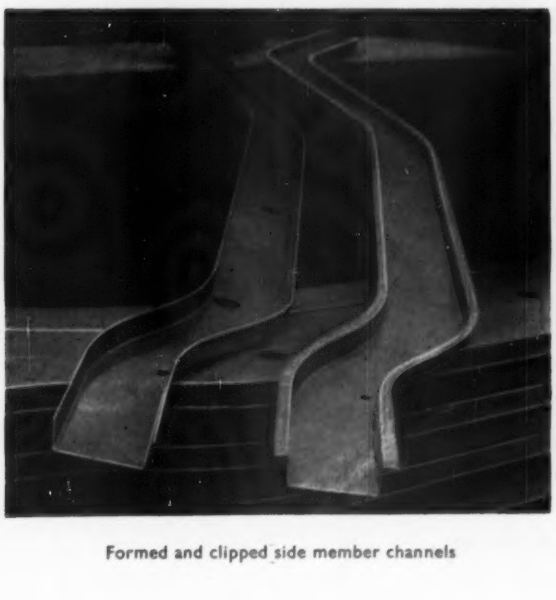
As required, sheets are cut to blank size on guillotine shears. When sheets are required to be shaped for economy of blanking scrap, they are marked on the sheet from templates and cut on rotary shears.

Sub-assembly

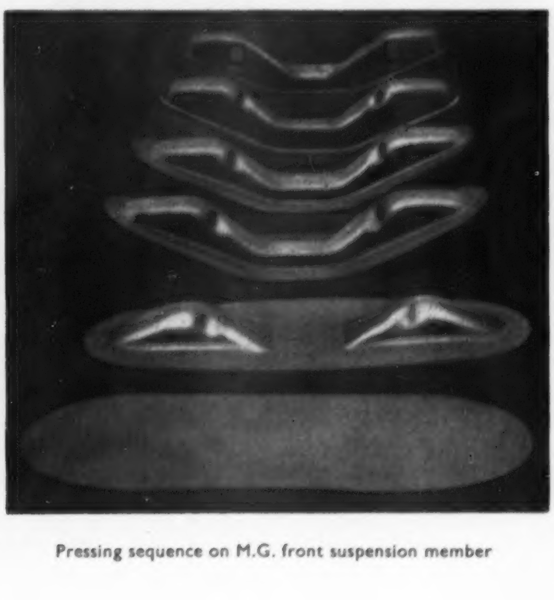
From the press shops component pressings are transferred to the assembly shop where various supplementary operations are performed to prepare them for final assembly. In some instances lamination plates or other stiffeners are spot-welded where necessary to the inner surfaces of the vertical webs of side members, and channels may be welded together to form box sections. Closing plates are welded to some channel-section side or cross members, and other components may be spot-welded or projection-welded in position.

Particular interest attaches to the Rover side member which comprises a pair of 16 S.W.G. channels, with narrow-lipped flanges, upswept at both ends and stiffened by a lamination plate at the front upsweep. After the lamination plates are spot-welded in position, each pair of channels is loosely assembled, flanges to flanges, over six "Stillite" sound-damping blocks, each approximately 6 in long and 3½ in thick. Spaced at predetermined positions, these blocks are slightly compressed between the flanges and webs of the channels and serve to locate the two pressings relatively and maintain them in a parallel spaced relationship. Such pairs of channels are then mounted between jig plates in a roll-over fixture and clamped by means of air cylinders to compress the "Stillite" blocks and bring the flanges of the two channels into contact with each other. They are then tack-welded at the flange lips to secure; first along one side and then the fixture is rolled over and the operation repeated on the other side.

From this fixture the tacked side members are passed to a carbon arc welding machine, set up between steel jig plates which are profiled to the contour of the upper run of the member, and clamped by a series of air cylinders. Flux is smeared over the



Formed and clipped side member channels



Pressing sequence on M.G. front suspension member

Rotatable frame welding jig

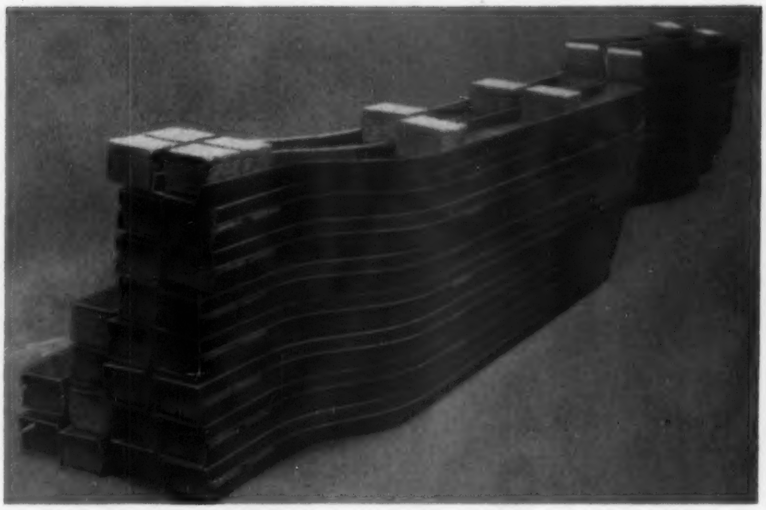
Rover frame on checking jig

Cross member welding jig

Welding up a Rover frame

abutting lips of the flanges and, when the machine is switched into operation, the table carrying the jigged side member is moved horizontally at a constant speed beneath a welding head arranged for vertical movement on stationary column. The head is counterbalanced, with a predetermined bias towards the work carriage, and a tracer needle fitted to the head runs on the profiled edge of the jig plate to maintain a constant arc length from the carbon pencil to the work. The weld is obtained by fusing together the adjoining flange lips, and the carbon pencil is burned away at a rate of approximately only ½ in to 10 ft of weld. Thus the welding operation is virtually automatic and requires only a minimum of supervision. On completion of the run, the side member is unloaded, transferred to a similar special-purpose machine having a jigging fixture profiled to the lower run of the member, and welded on the underside.

A Daimler side member is formed of two 12 S.W.G. channel pressings, fitted one inside the other to make a box section and hand welded top and bottom. Another boxed side member, for an Alvis frame, consists of a 14 S.W.G. top-hat section closed by a bottom plate of the same gauge spot-welded to the side flanges.



Rover side channels assembled with "Stillite" blocks prior to welding

Assembly

It has already been noted that present day passenger car frames are invariably of welded construction. While a welded frame has numerous functional advantages, not least the possibility of weight reduction, it has introduced some manufacturing problems. It will be appreciated that it is easier to achieve dimensional accuracy in a riveted frame than in a relatively complex welded construction. A minor divergence in the set of a riveted component can be readily detected before final assembly and can probably be individually corrected, whereas a discrepancy in a welded component may not become apparent until it has been welded into the structure. A careful study of each frame design is necessary to ensure regular production of a welded frame, to the close limits specified, requiring the minimum of correction and hand setting after fabrication.

The frame is broken down into as many sub-assemblies as is practicable and each side member or cross member has its individual welding jig. As many details as possible are welded in at this stage before the component is passed to the final assembly welding jig.

It is probably inevitable that distortion will occur in a welded fabrication. Its extent is not predictable with accuracy but much can be done to minimize and control it by careful design of jigs and fixtures, by adopting an appropriate welding technique and by regularizing operational procedure. Each sub-assembly and each final assembly has to be assessed in accordance with the amount of weld metal to be deposited. Commonly, a certain presetting of component parts is necessary in order to obtain a correct result. Wherever possible, of course, such presetting is incorporated in the press tools and must be proved by trial assembly. In the absence of such pre-

Main bay of toolmaking shop

Automatic welding machine for Rover side members

cautionary measures prior to assembly it is possible for the overall length of a frame to be reduced by as much as $\frac{3}{16}$ in as a result of the welding operations. Satisfactory production cannot be obtained solely by reliance on the jigging fixture to hold the component parts accurately in position. If such a method is attempted it may be impossible, after welding up the frame, to remove it from the jig.

Assembly welding jigs are built up on a single central girder, with location pillars and toggle clamps mounted on cross bearers. Shielded in heavily curtained, individual booths, the jigs are rotatable through 360 deg, with positively locked intermediate stations, so that the work can be manipulated to convenient position for the teams of welding operators. All the welding cannot be performed whilst the frame is mounted on the jig owing to interference by locating and clamping devices and the welding is completed while the frame rests on a table support.

After the final welding, and such minor operations as the jig drilling of a few holes not pierced in the pressings, the frame is cleaned free of all weld slag and then passed to a checking fixture. On this, such items as engine, suspension, and body mountings, steering supports and shackle tubes are checked for accuracy of position to various tolerances of from 0·030 in to 0·015 in and, when necessary, corrected by manual setting. From datum faces on the fixture, the angularity or parallelism of mounting surfaces is checked by metal set squares and heights or depths by adjustable gauge blocks. All welding is carefully examined and any deficiencies are marked for immediate rectification.

On passing the final inspection the frame is transferred to the finishing shop and placed on a conveyor to be passed through the degreasing plant. This is a cabinet housing two banks of jets through which is sprayed under pressure a mixture of degreasing solution in the first chamber followed by a hot-water spray in the second chamber. The spray in each case is almost at boiling point, and when the frame emerges from the cabinet it is virtually dry. After dipping in a tank of black enamel it is suspended on the conveyor of an 18 ft wide, tunnel-type, oil-heated stove. The passage of the work through the draining section, the stoving section maintained at a temperature of 450 deg F, and the cooling section, occupies about 40 minutes. From the finishing shop the frames are transferred to the dispatch bay and stacked ready for transport, generally by road, to the manufacturer.

The extensive works of John Thompson Motor Pressings Ltd. have a roofed area of 305,000 square feet. In conjunction with associated companies of the John Thompson Ltd. group in the immediate vicinity, it is served with hydraulic power, compressed air for shop lines, and water at 340 deg F and 170 lb/in^2 for shop heating, from a central power station. For both passenger cars and heavy commercial vehicles, more than 150,000 pressings are produced each week.

Tunnel-type, oil-fired enamelling stove

TRACK-TYPE TRACTOR TRANSMISSIONS

CAREFUL matching of tractor speeds to working requirements is necessary to ensure efficient and low cost operation of track-type tractors. An example of how such matching may be effected is provided by the 150 h.p. Caterpillar D8 tractor, which is available with either a standard transmission or one of two optional transmissions. Bulldozing, log-hauling and push-load scraping are some of the many duties for which the standard transmission giving five forward speeds (1·9 to 5·8 m.p.h.) and three reverse speeds (1·9 to 5·8 m.p.h.) is suitable.

One optional transmission having five forward speeds (1·4 to 5·0 m.p.h.) and three reverse speeds (1·9 to 5·2 m.p.h.) enables the tractor to exert 39,800 lb pushing effort at 1·4 m.p.h. for loading wheel-tractor drawn scrapers. Sand can be pump-loaded into the scraper in 2nd gear. Production bulldozing, with a straight or angled blade, can be accomplished in 2nd gear with ample reserve power in 1st gear when stubborn loads are encountered.

The second optional transmission is basically the standard transmission with different gears substituted to raise the 4th and 5th forward gear speeds to 5·2 and 7·2 m.p.h. and the 3rd reverse gear speed to 7·2 m.p.h. This transmission is particularly applicable to logging operations.

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

A QUICK-ACTION jig clamp developed by Speed Tools Ltd., Vereker Buildings, Gresse Street, London, W.1, is illustrated in Fig. 1. It incorporates a combined cam and screw action that provides quick locking with positive clamping, which is obtained by means of a forked cam pivoting on a sleeve nut. Thus quick clamping is obtained by means of the cam, which reaches a locked position with the handle conveniently inclined at an angle of 15 deg. Additional pressure can then be brought to bear by using the handle as a tommy bar and tightening the sleeve nut on to the stud. To release the clamp it is merely necessary to unlock the cam, and the clamping arm is then lifted clear of the work by means of an integral spring.

The clamp strap is a medium carbon forging, machined on all working faces. The cam unit, of cast steel machined from the bar, is hardened and ground on all faces. It bears on a stout washer of medium carbon steel, which is also hardened and ground. At the lower end of the steel sleeve there is a shoulder to retain a supporting spring. By this means a constant pressure at any height is maintained, and the necessity for making special springs, as with other forms of strap clamping, is eliminated. The steel handle is cyanide hardened and fitted with a plastic ball knob for comfort of operation. The studs and heel pins are of high tensile steel and are provided with lock nuts. With the handle removed, the clamp unit takes up no more space than a strap clamp with the conventional nut for spanner tightening. It will, however, be appreciated that a large saving may be effected by eliminating the need for spanners.

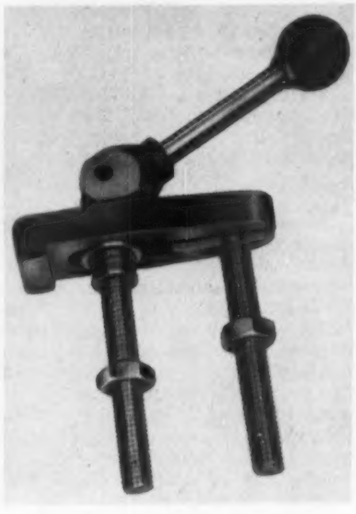


Fig. 1. Speetol jig clamp

These jig clamps are made in a range of three sizes to suit $\frac{5}{16}$, $\frac{3}{8}$ and $\frac{1}{2}$ in Whitworth studs. Studs and heel pins are provided with each unit. They are of sufficient length to allow for all normal requirements. Where necessary, extra long studs may be fitted by the user to suit any height that is to be clamped.

Grinding machines

Two grinding machines recently developed by The Gardner Machine Co. of the United States of America are illustrated in Figs. 2 and 3. The machine shown in Fig. 2 is a special high-production two-headed grinder for grinding the closed end of automobile valve tappets. It has a heavy welded steel base, supporting two grinding head slides upon dovetailed ways. Ease of movement is ensured by employing hardened and ground steel plates to serve as races for precision ball bearings. Each grinding head thus operates on ball bearing ways.

Each spindle is 3·937 in diameter in the bearings at the wheel collar end. A double roller type bearing takes the radial load and two ball thrust bearings carry the thrust load. Each head is arranged to pivot to allow angular adjustment of the grinding wheel. Automatic feed is provided for each grinding head slide, but these slides can also be operated manually. A caliper on each head registers from the ground surface when insufficient stock is being removed, in which case the head is automatically moved in a predetermined amount. Counterweights hold the head against adjust-

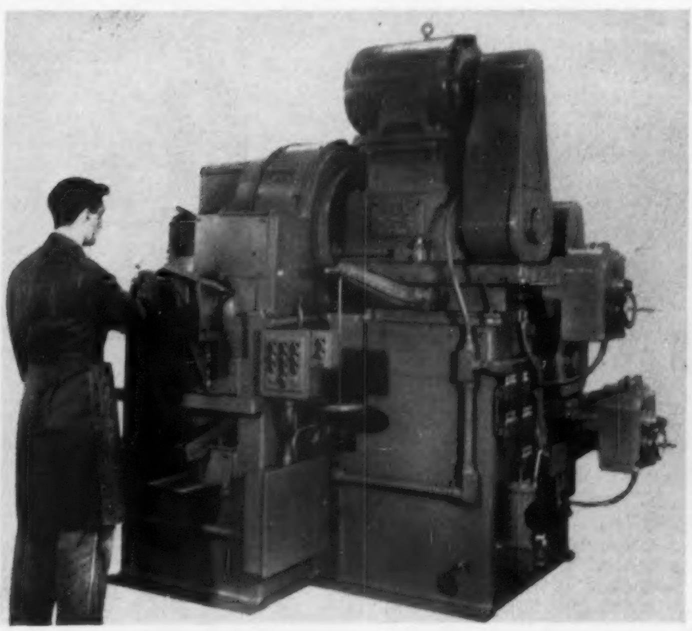


Fig. 2. Gardner horizontal spindle two-head grinder

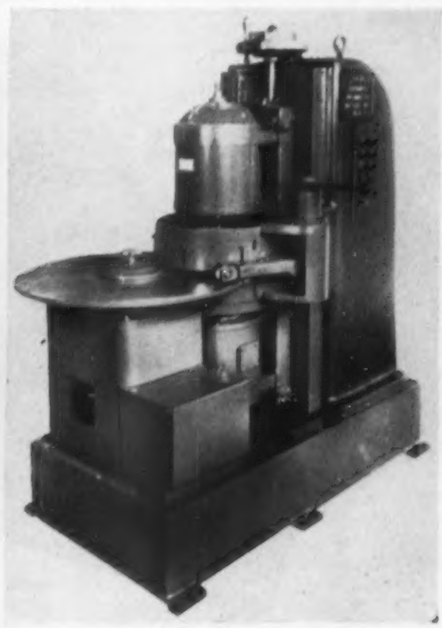


Fig. 3. Gardner vertical double-spindle grinder

able stops. There is a power-operated swinging dresser, employing ball bearing cutters, for each head. The dressers are enclosed within the head and are mounted at the rear.

A rotary work carrier of the vee notched type is operated through a Vari-speed unit that permits speeds within a 3:1 ratio. A chain "hold-down" attachment keeps the tappets in the carrier vees while they are being ground. An inclined trough into which the tappets are stacked permits the work carrier to pick up the work pieces one at a time as it revolves. Unloading is by gravity. With maximum stock removal within the range 0.028-0·038 in, production is at the rate of 2,400 tappets per hour within 0·002 in for uniformity.

The Gardner No. 2V-18 vertical double spindle grinder, shown in Fig. 3, is built to grind simultaneously, two parallel faces of small work pieces. It is particularly suitable for grinding coil springs, carbon brushes, ceramic materials and similar small parts. The work pieces are loaded by hand into openings of a circular work carrier that is mounted horizontally and power driven. A quick-acting screw release on the centre clamping plate makes it possible to effect carrier change-over very quickly.

The machine column is a heavy iron casting that provides great rigidity. The work carrier drive embodies a gear-type speed reducer with a 434·7:1 speed reduction. By finger tip adjustment of a variable speed hydraulic mechanism, speeds from ¼ to 1 r.p.m. are obtained from the work carrier. Two 18 in diameter Gardner Wire-Lokt abrasive discs are employed. They are driven by 5 h.p. totally enclosed motors. As these grinding discs are solid, with no central hole, the work passes directly across the centre of the wheels. This provides an effective diameter that is larger than normal, and ensures maximum wheel economy and less frequent wheel dressing. A hand-operated swinging arm dresser may be seen at the right of the illustration. It is a sturdy device, with the pivot shaft fully enclosed and protected against the dust and grit of grinding. It is a remarkably efficient truing device. If desired, a wet grinding system can be supplied. Burton, Griffiths and Co. Ltd., Mackadown Lane, Kitts Green, Birmingham 33, are the sole agents for these machines in Great Britain.

Gear tester

The first of a range of similar machines to be built by The David Brown Tool Company of Huddersfield, Yorkshire, is illustrated in Figs. 4 and 5. It is designated the David Brown No. 24 gear roll tester, and is designed to provide a simple, rapid and positive means of checking concentricity, tooth contact, centre distance and varying tooth thickness of gears up to 24 in. External or internal spur and helical gears are accommodated on the standard model shown in Fig. 4, while separate heads, mounted on the adjustable carriage, see Fig. 5, are available for shaft gears, bevel gears and worm gears.

This is a machine of extreme precision and simplicity that is very well suited for repetition work (using a master gear) or for checking master gears themselves. Readings are registered on a dial indicator, sensitive to 0·0005 in, while centre distances are read directly from a vernier scale.

The No. 24 gear roll tester comprises three main components; the bed, the adjustable carriage and the sensitive carriage. To ensure the rigidity that is essential in a precision machine of this type, the bed is of ribbed construction and the top is provided with slideways that locate both adjustable and sensitive carriages. The adjustable carriage is located on two guideways, one of vee section and the other rectangular. It is moved by a handwheel that operates a pinion meshing with a rack. Provision is made for locking this carriage in any desired position along the bed. Location is provided on the standard bed for mounting the centre and bevel attachments. A No. 4 Morse taper locates the work arbor.

Four balls, mounted in hardened and ground guideways to ensure complete freedom of movement, locate the centre carriage. This slide operates against spring pressure, which can be varied by adjusting hand units. Spring pressure can be exerted in either direction, to allow for external and internal gears. An indicator pin shows in which direction the spring is acting. The dial indicator can be adjusted by screw or set at zero by rotating the dial. When the gear being checked is sufficiently large to cover the sensitive carriage, the indicator can be removed and placed in an auxiliary bracket to give the operator an uninterrupted view. A safety screw is fitted to prevent damage to the indicator by over-running of the sensitive carriage. A scale is attached to the sensitive carriage and a vernier is fitted to the adjustable carriage to provide accurate measurement of gear centre distances.

Three attachments greatly increase the range of work for which this gear roll tester can be used. There is a centre attachment arranged for mounting on the adjustable carriage. It will accommodate shaft gears up to 10 in maximum gear diameter, and is fitted with an adjustable centre mounted on

Fig. 4. David Brown standard gear roll tester

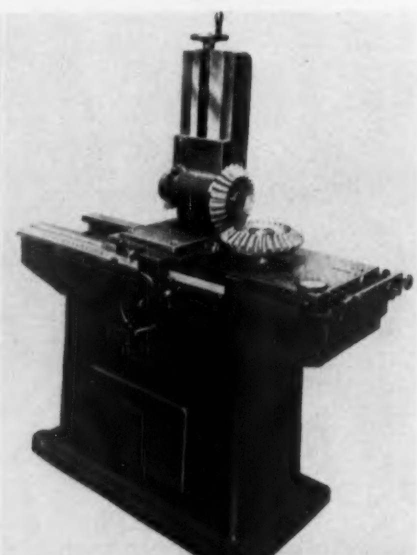


Fig. 5. Roll tester with bevel gear attachment

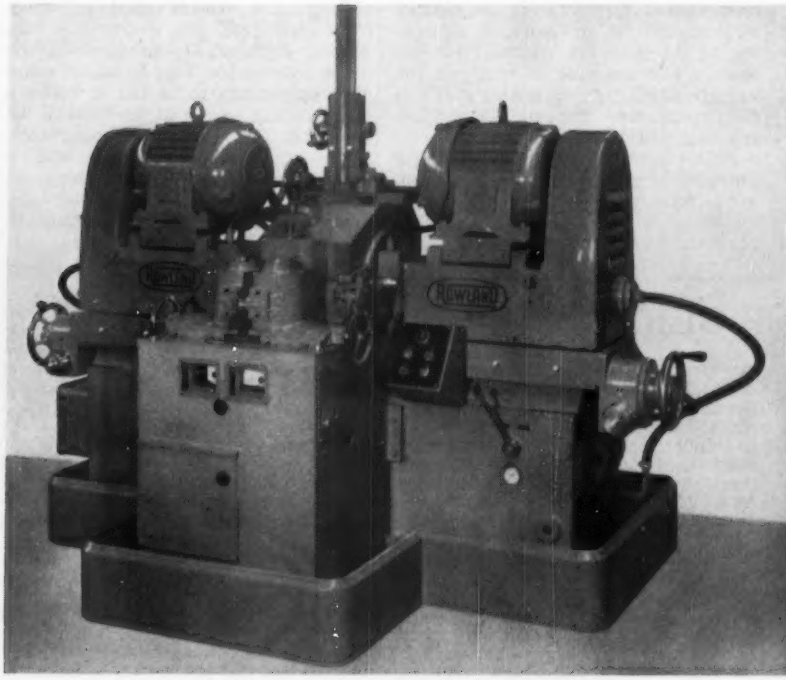


Fig. 6. Rowland 'through-feed' duplex surface grinding machine

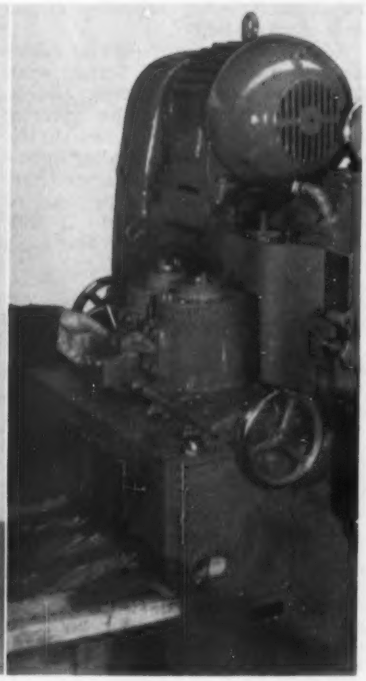


Fig. 7. Feed rollers on the Rowland grinder

a vertical sliding bracket. A centre is also located on the adjustable carriage, and shafts up to 20 in overall length can be mounted.

The bevel attachment, which is arranged for mounting as a separate unit of the adjustable carriage, will accommodate straight or spiral bevel gears. Apex distances are read from the vernier scale on the bevel attachment, which is set to zero from the face of the arbor locating bush on the sensitive carriage, and from the vernier set on the bed. The adjustable stop on the bed is used when quantities of bevel gears are to be tested.

Worms up to 10 in maximum diameter can be accommodated by the worm attachment, which is supported by the adjustable carriage. The worm support brackets are adjustable in order to locate shafts of varying lengths, which may be mounted in bushes or between centres. The horizontal slide carrying the brackets can be offset 5 deg either side of centre, the measurements being taken from a circular vernier scale. This makes it possible to determine the amount of correction or adjustment necessary to the worm wheel generating machine to obtain correct contact markings.

Gears are tested by rotating them together in pairs. When quantities of similar gears are to be tested it is recommended that one of the gears should be a master or one that is known to be correct. With the gears mounted on their respective arbors, the adjustable carriage is moved along the bed to bring the gears into mesh, and is then clamped in a position where it will exert a slight spring pressure against the sensitive carriage.

To ensure freedom of movement and to minimize the risk of wear, it is advisable to mount the gear of least weight on the sensitive carriage. When the gears are rotated any variation in concentricity or tooth thickness is immediately registered on the dial indicator, while the centre distance at which the gears are meshing is read on the vernier scale. Backlash can be calculated from the distance between the working centre distance and the centre distance when the gears are in metal-to-metal contact.

"Through feed" surface grinding

F. E. Rowland and Co. Ltd., Climax Works, Reddish, Stockport, who have been carrying out considerable development work in connection with their range of duplex surface grinders, have recently introduced a machine for accurately and simultaneously grinding the opposite parallel faces of square, rectangular and circular components. This machine is illustrated in Fig. 6. It is fitted with 20 in diameter grinding wheels, and allows continuous grinding by incorporating the "through feed" principle.

The machine bed, of substantial fabricated steel construction, is strongly ribbed and has ample base area. Cast iron slideways of large section are secured to the upper surface. They provide narrow guides for the workhead to ensure that accurate alignment is maintained. Each grinding wheelhead is driven through multiple vee belts by a standard totally enclosed 15 h.p. motor. The high tensile steel spindle of each wheelhead runs in precision pre-loaded angular contact ball bearings at the grinding wheel end, and in a roller bearing at the driving end. This construction ensures that increase in length due to temperature rise does not affect the accuracy of the work.

The abrasive discs, of the inserted nut type, are mounted on heavy steel backplates attached to the wheel spindle adaptor. They have been specially developed. Each wheelhead is universally adjustable so that the grinding wheels can be set relative to each other and to the plane of the feed of the workpieces to provide the most efficient grinding conditions. The wheelheads are held against dead stops by hydraulic pressure applied through a motor-driven pump and valve gear. To eliminate any possibility of contamination of the hydraulic fluid with coolant, the pump unit and reservoir are located outside the machine bed.

At the rear of each wheelhead there is a substantial feed unit to control the grinding size. These units may be fitted with an electro-hydraulic device whereby variable increments of feed can be applied to compensate for wheel wear. This device is push-button controlled from the operating position. Feed to the workpieces is effected by the pressure of two contra-rotating rubber-covered feed rollers, which are adjustable for width to allow components of various thicknesses to be passed through the machine. Drive to the feed rollers is provided by a foot-mounted motor through a vee belt and worm reduction gears. The speed of the rollers is steplessly variable within certain limits. The whole of the feed roller assembly is mounted at the front of

the machine in an extension of the main machine bed.

Adjustable entrance and exit guides are provided. Top and bottom work guides, passing fully across and in between the faces of the grinding wheels from front to rear, restrain the workpieces during the grinding operation. These guides are independently adjustable, and allow a wide range of sizes to be accommodated. A substantial wheel dressing device is incorporated. Dressing may be carried out with cutters or diamonds, according to the accuracy specified for the work. The machine is arranged for wet grinding, and the coolant is fed to the workpieces through the hollow grinding wheel spindles.

Push-button operated controls are provided for the hydraulic pump, work feed and grinding wheelhead motors. A warning light is incorporated, which is illuminated when the grinding wheels need replacing.

In operation the workpieces are loaded continuously in between the work rollers, see Fig. 7, which cause the components to be fed in between the grinding wheels at the front of the machine. The work passes completely across the grinding wheels and is ejected at the rear of the machine. A similar type of machine in a 30 in size is also available for dealing with components of larger dimensions.

RECENT GERMAN PUBLICATIONS

Small Gas Turbines especially for Vehicle Propulsion, (Kleingasturbinen insbesondere zum Fahrzengantrieb)

By Prof. Dr.-Ing. K. Leist.

Cologne and Opladen: Westdeutscher Verlag, 1954, $11\frac{3}{4} \times 8\frac{1}{4}$. 104 pp. 85 illustrations. Prime DM.22 (1 DM = 1s. 8d.).

This is one of a series of research reports published by the Ministry for Economics and Transportation of the Country of Northrhine-Westfalia. The author, who is the head of the institute for turbo-machines at the Technical High-school at Aachen, is well qualified to deal with the subject. The present volume sets out to consider the suitability of gas turbines for relatively small outputs as required for lorries, tractors, omnibuses and cars, a problem which requires clear headed non-partisan consideration, particularly in view of immense gas turbine successes in other fields and against, or rather in spite of, at times an unhealthy optimism displayed by some over-enthusiastic exponents of this new art.

The first 23 pages are devoted to general turbine configurations which in view of the author's aim to deal with fundamental considerations follows along conventional lines. The second chapter (7 pp) deals with losses encountered particularly because of the small size of turbines considered. In support of his conclusions, the author cites some data concerning the effect of Reynolds number as estimated previously by Barr and vehicle load factors arrived at on the basis of Centrax estimates. The important subject of small gas turbine economics is considered next (21 pp). Admitting that because of the fuel consumption the economic balance is against the turbine, the author endeavours to make a case on the basis of the calorific values required by the gas turbine versus the reciprocating engine and the possibility of using bunker oil for the former which should reduce fuel costs, whilst the reduced weight might balance the weight due to required increase in fuel capacity of the vehicle. Following this argument, the author turns to the possibility of using pulverized coal, a development which lately has been actively pursued in the U.S., Canada and this country, although only U.S. data is referred to. The author puts forward "am Rande" what he himself terms a somewhat "kühne" suggestion according to which the turbine rotor should be driven direct by the gases emerging from a coal burning combustion chamber.

No chapter on gas turbine economics is complete without reference to heat exchangers and here the author does not make an unduly great virtue of what is seldom an attractive necessity, although he draws attention to the additional benefits due to permissible lower compression ratios. The well known attractions of the gas turbine torque characteristics are next considered, and the next section, devoted to weight and space requirements, is in the main based on the well known Boeing powered Kenworth lorry supplemented by results of a paper study by the M.A.N. In a chapter headed "Further considerations," the author mentions noise problems which he hopes will be amenable to a satisfactory solution and then lists possible advantages of the gas turbine: reduced wear, smoother running, reduced lubricating oil consumption, simpler ignition system, no water cooling system, reduced number of parts, easier cold starting, no detonation problems. This chapter ing, no detonation problems.

The final chapter is devoted to brief descriptions and performance data of such well known units as the Boeing 502, Rover, Turbomeca and Solar units as well as the Budworth turbine. In addition the Centrax, the French Socema and Laffly and the Spanish C.E.T.A. turbines are described as well and a brief summary is signed by the author and Dipl.-Ing. K. Graf.

It will be gathered that the book is well suited as an objective general introduction to the subject. Its value would be further enhanced by devoting more space to the all-important problem, for vehicles, of part load performance which has not as yet received the attention due to it. The description of contemporary small gas turbines could be brought more up-to-date by reference to the W. H. Allen and the Fiat turbines, whilst the argument in favour of coal burning units would benefit from a reference to the North-British Parsons locomotive. The book is meniographed and paper bound and the price of almost £2 seems to be somewhat excessive for a production of this type.

Vibrations of the Motor Car and Engines (Schwingungen des Kraftfahrzeuges und der Motoren)

By E. A. Wedemayer.

Berlin, W.35: Technischer Verlag Herbert Cram. 1955. $6\frac{3}{4} \times 9\frac{1}{2}$. 126 pp. 195 illustrations. Price DM.24 (1 DM = 1s. 8d.).

The purpose of the book is to serve as introduction and guide to specialists as well as students who have to deal with or learn the fundamentals of vehicle and engine vibrations. Both subjects are dealt with in one volume since they are likely to influence each other and in turn the riding qualities of the vehicle. The book is divided into three parts, the first (31 pp) dealing with fundamentals of mechanical vibrations. In this section the performance of vibration dampers, non-linear systems and systems with more than one degree of freedom benefit from a brief but clear and lucid treatment. The second part (37 pp) is devoted to vehicle vibrations. In it the author endeavours to deal only too briefly with such complex subjects as road irregularities, effect of vibration on passengers, tyre performance, vibration dampers, independent suspension, vehicle pitch and bounce vibrations and their experimental determination, shimmy, motor cycle suspension, trailer couplings, trailer instability and brake surge. Considering the complexity of the matter dealt with under these headings, it is obvious that apart from mentioning the main aspects (and that not in every instance) the author could not do justice to the subject within the limited space. This is particularly apparent in connection with the treatment of vibration sensitivity of passengers and vibration dampers which ignores the comprehensive studies due to Lebr and his concrete conclusions for dealing with vehicle bounce and pitch vibrations; the work is unduly descriptive and again ignores the work due to Lebr and Bertschinger, Olley and Slaby to name but a few, whilst the pages dealing with the experimental aspect of the subject are rather superficial, particularly against the background of more recent papers on the subject. Again, the chapter on trailer couplings would benefit from a consideration of the ring spring, whilst no treatment of trailer instability is complete without reference to Williams' paper on the "Mathematical Theory of the Snaking of Two-Wheeled Trailers" (*Proc. Inst. Mech. Eng. Auto. Div.*, 1951-52, Part IV).

The last part deals with engine vibrations and here balancing is considered in some detail along conventional lines, torsional stiffness of crankshafts being in the main dealt with according to the data due to B. C. Carter, no reference being made to the more recent investigations due to B.I.C.E.R.A. The inter-action between wheels, brakes and engine in terms of torsional vibrations is considered next and the book is concluded with a review of camshaft vibration problems and two pages devoted to noise.

The Bibliography includes 308 references mostly of pre-war vintage and here it is curious to find reference to Timoshenko and Lessels "Applied Elasticity" in English, although a German translation was published in 1928, whilst no mention is made of Cain's book on *Vibration of Rail and Road Vehicles*, or Marquard's book on *Vibration Dynamics of Fast Road Vehicles* (reviewed in our issue for March 1953), both of which relate closely to the field covered by the present volume. Nevertheless, the book is both interesting and stimulating but it is rather obvious that the very large field dealt with could not be adequately covered within a small volume.

PRECISION ALLOY STEEL CASTING

A Process with Great Potentialities

IN 1949, B.S.A. Tools Ltd. developed a process for the precision casting of milling cutters, in which the resulting metallurgical structure and mechanical properties very closely resembled those of the conventional types of high-speed steel. The immediate advantage was a considerable reduction, in the region of 20 per cent, in the cost of producing milling cutters. In addition to this and of particular importance was the fact that the casting process relied on the use of a steel in which chromium was the chief alloying element instead of tungsten or molybdenum, which were in very short supply at that time. Of even greater value to the tool designer was the freedom of design that the casting process offered, since he was no longer restricted by conventional methods of machining from the solid. Staggered-tooth cutters of improved design and superior cutting performance could be cast as easily as the straight tooth variety and both could be produced more cheaply than by conventional machining methods. Typical cast cutters produced by this process are illustrated in Fig. 1.

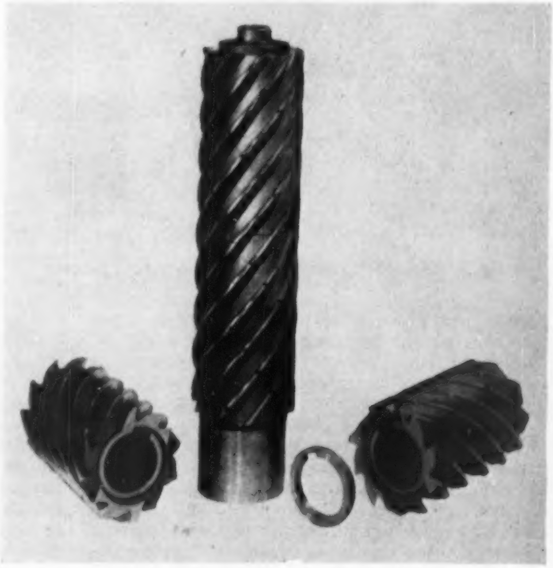


Fig. 1. Cast cutters produced by the P.A.C. process

Cast cutters proved so successful that it was decided to apply the process to other articles such as forging, extrusion and plastic moulding dies and general engineering components. Further development work carried out by B.S.A. Tools Ltd. showed that the process was adaptable to almost any steel, and that practically any required combination of mechanical properties could be obtained. Considerable interest was aroused among customers who co-operated during the initial development period and it was soon evident that to exploit the process on a sound commercial basis, a separate company would have to be formed. On August 1, 1954, Precision Alloy Castings (Birmingham) Limited officially came into being for the purpose of developing and applying the original B.S.A. Tools precision casting process to a much wider field. Already the new company is handling a large number of products for the jig, tool and die industry in addition to the general engineering industries. Designers are rapidly revising their outlook towards casting because this new development has distinct possibilities for many applications previously considered unsuitable for production by conventional casting techniques.

Advantages of the process

Almost any type of steel may be cast by the process and practically any combination of mechanical properties can be obtained. With this in mind, it is easy to assess the value of the P.A.C. process from a purely practical point of view. Primarily, the process enables any tool die or component to be cast very close to its final shape; even for components of high precision it is necessary to leave only a small amount of material for finish machining. It will be appreciated that a considerable saving in the total cost of production can be achieved since there is immediately a reduction both in machining time and the use of machine tools. Rough machining is eliminated altogether, and the plant normally engaged on this operation can be re-allocated to finish machining —virtually doubling the capacity of the toolroom. Furthermore, the skilled toolmaker may concentrate on the most important part of his job, namely, that of finishing the tool or die. For some components, such as certain types of plastic moulding dies, the forms may be cast so precisely that only bench finishing or polishing is required. An additional saving is in the cost of raw material, particularly when, as with most orthodox methods, a lot of metal has to be removed which is normally wasted. A further advantage of the process is that P.A.C. castings have been found to give an equivalent or even better life than items produced by more conventional methods. This has been particularly noticeable with extrusion dies for aluminium, brass, copper and steel, and also with stamping dies where a life at least as good as that obtained from dies made by conventional methods has been obtained when producing stampings in a 13 per cent chromium steel to S62.

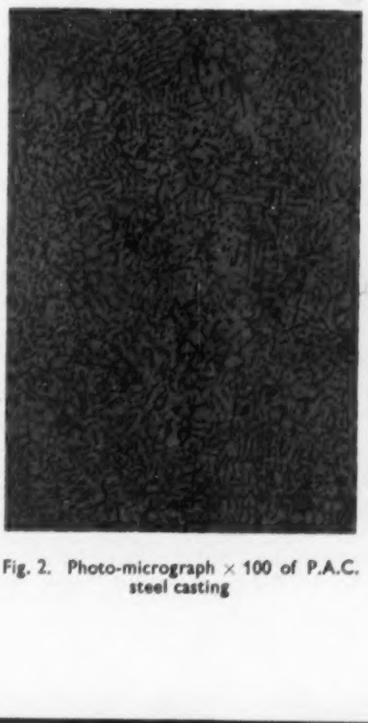


Fig. 2. Photo-micrograph × 100 of P.A.C. steel casting

Since castings can be produced from comparatively low priced patterns, the process is equally economical for either large or small numbers of components and very little capital outlay is involved. There is almost no limitation to the type of steel that can be cast, and in fact about forty different types of alloy steels are being produced at the moment. These include case hardening, constructional, carbon and alloy tool steel and heat and creep resisting types. In addition, it is possible to obtain practically any mechanical properties required by the relevant B.S.I. and D.T.D. specifications. Special combinations of mechanical properties can be obtained, and when required a customer's own specification can be met. Conventional heat-treatment can be applied to products cast by this process and components or tools can be supplied in either the heat-treated or fully-softened condition.

To complete the list of advantages of this process a word regarding accuracy

is necessary. In this respect a limit of approximately ±2 per cent can be maintained on important dimensions as a general rule, but this figure depends on both size and complexity. In certain cases it may be possible to improve on these figures.

Structural and mechanical properties

P.A.C. steel castings comply with the most stringent conditions for general structure and soundness. The structure of the material shows a good, even, small grain size, see Fig. 2. For most commercial applications with open tolerances, the surface finish of P.A.C. castings is good enough to obviate the need for finish machining operations, see Fig. 3. Fine tolerances, of course, will necessitate some finish machining, but this can be reduced to a minimum with good design. To the customer, a further guarantee of soundness and quality lies in the facilities existing in the P.A.C. inspection department, which is equipped to carry out mechanical testing, supersonic and radiographic examination as well as accurate measurement of dimensions and surface finish. The latest supersonic and radiographic equipment is used to examine all, or a proportion of the castings as required by the customer. A mechanical test certificate and prints of the radiographic tests will be supplied on request.

Whatever method is used to manufacture a component, the designer's main concern is that the component shall have the ability to stand up to the stresses and strains to which it will be subjected during its working life. This ability is judged largely on the mechanical properties exhibited by a test piece of standard dimensions when subjected to static tests. Owing to the inherent nature of steel, these properties depend on the size of the component. This has been taken into account by P.A.C. and a whole range of steels with increasing quantities of alloys are used so that uniform mechanical properties may be obtained throughout a component by using the correct steel.

Fig. 3. Engineering components cast by P.A.C. process

An important feature of these castings is that the desired mechanical properties can be obtained through conventional methods of heat-treatment such as hardening, tempering, normalizing and annealing. Castings can be produced which will meet the mechanical requirements of all the B.S.I. and D.T.D. specifications for castings, and in some cases closely approach the properties called for in the B.S.I. and D.T.D. specifications for steel in the wrought condition. Furthermore, the flexibility of the process often enables a special combination of mechanical properties, not conforming to a standard specification, to be obtained to designer's own requirements. Castings produced by P.A.C. can be supplied in the heat-treated condition if required, such heat-treatment being carried out in the P.A.C. works.

The field of application for alloy steel castings is constantly expanding and as fresh applications are found, its scope will be extended still further. Already P.A.C. castings are being extensively used in the jig, tool and die industry; extrusion dies, hot stamping dies, plastic moulding dies and pressure die-casting dies, have been produced very satisfactorily which, on production, are achieving remarkable economy. In the general engineering industry, for such items as tools, tool-holders, impeller blades, machine tool parts, and many small and intricate components, the P.A.C. process is also being used to advantage.

Design

The precision alloy casting process undoubtedly presents vast possibilities and is yet only on the fringe of possible fields of application. In the foundry, technical advances will continue to broaden the scope of the process but another factor will in great measure determine the extent to which it can be put to practical use. This is simply the degree of understanding and collaboration that is established between the designer and the founder. It is essential for the designer to appreciate in some measure the technicalities of the process and the special difficulties that face the founder. Most of these difficulties can be overcome quite simply by the right approach to the design of the product. By allowing the founder as much freedom as possible in modifications to the design (where these are necessary) and by co-operating fully with him, the designer will be assured of a first class product and will contribute greatly to the overall development of the process. To sum up, it is important that each job be treated on its own merits. The advice of P.A.C. technical staff, together with development facilities, is readily available to assist anyone wishing to investigate the use of precision alloy steel castings. Whilst progress is being made in the casting of steel, other possibilities are also being investigated, such as the casting of non-ferrous alloys based on copper and aluminium. Increased efforts are also being directed to the aircraft industry, where a successful application of precision alloy castings has already shown that many outstanding problems at present being experienced in the specialized machining of high precision components could be overcome by the process.

APPLYING SEMI-SOLID ANTI-CORROSIVE COMPOUNDS

THE Regent Oil Co. Ltd. and the Atlas Diesel Co. Ltd. have recently co-operated in developing a hot spray method for applying semi-solid anti-corrosive compounds. Hitherto it has been found difficult to spray such compounds effectively, and more often than not excessive quantities of thinners were required, and consequent evaporation after application could result in an inadequate coating as well as contaminating the atmosphere with solvent vapours.

Another and normal alternative is to apply the compound with a brush. This method is time consuming and often quantitatively wasteful. It also gives an uneven finish and insufficient coverage in spots not easily accessible with a brush.

The newly developed "hot spray process" is the result of experiments with the Regent Oil Company's Caltex rustproof compound L and the Atlas Diesel Company's KV3 heater and Ecco 306 spray gun. The compound, which contains 12 per cent volatile thinner, is heated in the cup heater to 140-160 deg F; the melting point is 120-125 deg F. When the thinner has evaporated after application the melting point is 145 deg F. It is sprayed at a pressure of 45 lb/in^2 by the special internal mix Ecco 306 gun which gives the required atomization. When the globules strike the surface being sprayed, they congeal immediately, thus eliminating wasteful back spray and giving an even coating.

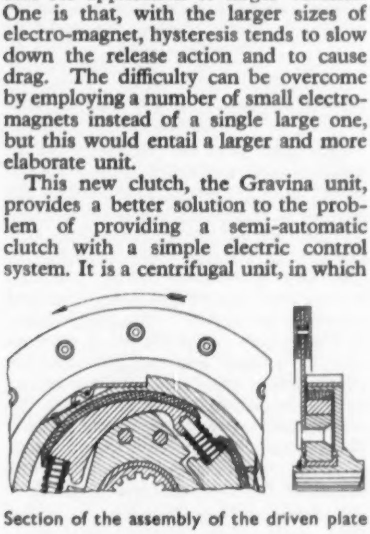
Compound L is a blend of mineral oils and petroleum jelly with special additives, which give it non-drying, flexible and adhesive qualities. None of these qualities is lost by the hot spray method.

THE GRAVINA CLUTCH

An Electro-Magnetically Controlled Centrifugal Unit

ALTHOUGH its electrical system is simpler, the Gravina clutch, made by S.A.F. Ferodo, Avenue de la Grande Armée, 64, Paris, is rather more complex than the Ferlec unit, described in the February 1955 issue of *Automobile Engineer*. For a number of reasons, a magnetic clutch of the Ferlec type is not altogether suitable for application to larger vehicles. One is that, with the larger sizes of electro-magnet, hysteresis tends to slow down the release action and to cause drag. The difficulty can be overcome by employing a number of small electro-magnets instead of a single large one, but this would entail a larger and more elaborate unit.

This new clutch, the Gravina unit, provides a better solution to the problem of providing a semi-automatic clutch with a simple electric control system. It is a centrifugal unit, in which



Section of the assembly of the driven plate

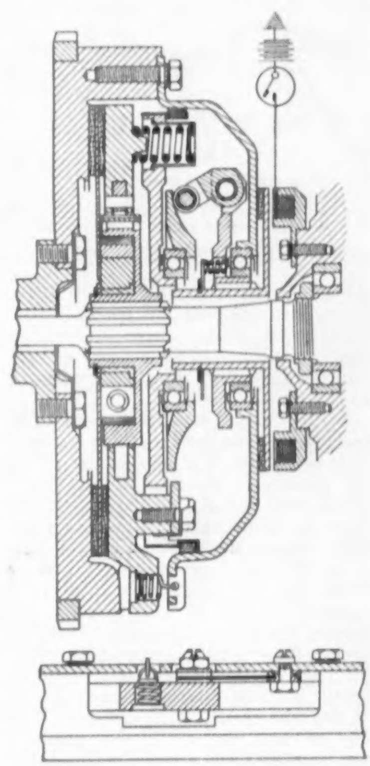
the electro-magnetic control is used to disengage a light, friction clutch that drives a splined sleeve on which is mounted the carrier of the centrifugal-weight assembly. The electro-magnet also brakes this assembly to stop it rotating. Since the sleeve is relatively light and moves axially only a short distance, a small electro-magnet is adequate. Therefore, hysteresis problems do not arise. Another advantage of this arrangement is that the consumption of electricity is small and only takes place during the movement of the lever to change gear.

The flywheel is cupped to house the clutch unit, and a steel cover is bolted to its rear face. The driven plate is clamped in the usual manner between the rear of the flywheel and a presser plate. It is faced with friction linings of conventional form, which are radially grooved to assist cooling. The plate is not splined directly on to the primary shaft, but has a special hub riveted to it.

Three cams are machined on the outer periphery of the hub, which fits into a drum, not unlike a brake drum, that is splined on to the primary shaft. The hub and plate assembly floats on a bush round the central boss of the drum. Housed inside this drum, between its peripheral track and the cam hub, are three, friction-faced shoes spaced apart by compression springs, which are interposed between them. The inner faces of the shoes are so shaped that they ride up the cams when the driven plate is turning in the normal direction of rotation. This action presses the shoes firmly against the track in the drum.

When the vehicle is over-running the engine, the drum tends to rotate faster than the driven plate, so the shoes automatically slide back free from the cam and slip takes place. This arrangement prevents the sudden braking of the vehicle in the event of the clutch being engaged with low gear selected when the vehicle is travelling too fast or the engine is not rotating fast enough. The compression in the springs that hold the shoes apart is regulated to such a value that, in these circumstances, the friction between the shoes and the drum is adequate to accelerate the engine to synchronize it with the transmission.

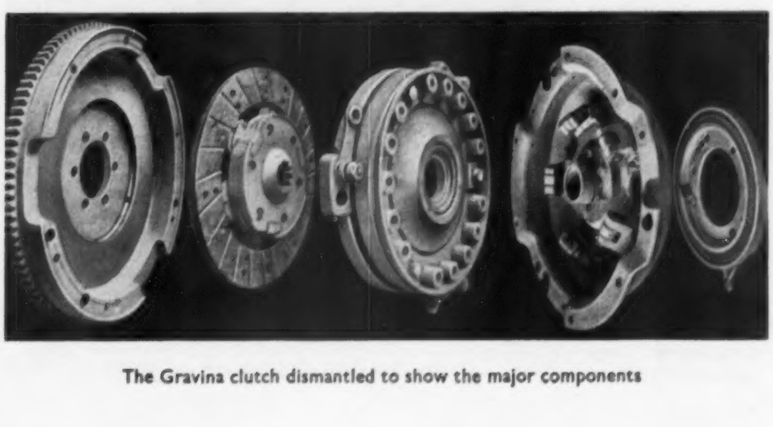
The presser plate is in the form of an annulus surrounding the hub of the driven plate. It is secured to the cover by three trailing links, of spring steel, tangentially disposed at intervals of 120 deg round its periphery. One end of each link is bolted to the cover and the other to a lug on the presser plate. The links transmit the drive to the presser plate but at the same time allow it a limited axial movement. Each of the three lugs also carries a return spring. The function of these springs is to withdraw the presser plate clear of the driven plate and thus to disengage the clutch.



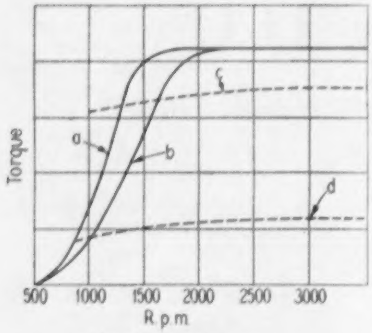
Cross section of the clutch in the engaged position

A groove in the inner periphery of the presser plate houses a number of pawls which, when the presser plate is either stationary or rotating very slowly, are forced by hairpin type springs to register in ratchet stops machined in the outer periphery of the drum round the driven plate hub. These pawls are so arranged that, when the vehicle is over-running the engine, they take the drive from the transmission to the power unit.

They have been incorporated for two reasons. One is to hold the vehicle stationary on a hill when the hand brake is inoperative. To do this, it is necessary to select reverse gear if the vehicle is facing up the slope and bottom gear if it is facing downhill. The other reason is to enable the vehicle to be tow-started, despite the fact that centrifugal force is not great enough to engage the clutch at low speeds. As soon as the engine has started and is turning faster than the primary shaft of the gearbox,



The Gravina clutch dismantled to show the major components



a—As the unit slows down
b—As it accelerates from rest
c—Engine torque
d—Engine braking torque during over-run

Torque transmission curves

the pawls ride over the ratchet stops. Then, at a relatively low speed, centrifugal force causes them to retract, against the action of the hairpin springs, into the groove in the inner periphery of the presser plate.

In the base of each ratchet stop slot, and pivoted near one end, there is a steel strip or tongue. The axes of the pivots are parallel to the axis of the drum. When the vehicle, and therefore the drum on the primary shaft, is stationary, these tongues lie flat in the base of their slots. They are held in this position by small leaf springs; one end of each spring is secured by a set screw to the base of the slot and the other bears up against the tongue end nearest the pivot. So long as the vehicle is in motion, centrifugal force overcomes the pressure of the leaf springs and lifts the other end of each tongue. In this position, the tongues form a ramp to mask the stop. Thus, in the event of engine stalling while a gear shift is being made, the pawls ride over the tongues, and are thus prevented from striking the stops and causing shock damage to the clutch, engine or transmission.

Immediately behind the presser plate is a reaction plate, against which the rear ends of the presser springs bear. These springs are housed in thimbles in holes round the periphery of the reaction plate. Three lugs, projecting from the rear of the presser plate, register in slots in the reaction plate to transmit the drive. A stop plate, bolted to the ends of the lugs after they have been passed through the slots, limits the axial movement of the reaction plate relative to the presser plate.

A cam ring, against which the centrifugal weights bear, is carried on a ball bearing mounted on a circular flange round the periphery of the hole in the centre of the reaction plate. It has the appearance of a plain ring with a fillet of large radius between its central boss and the periphery. The fillet forms the cam profile. When the weights bear against the cam ring, both the ring and the cam assembly tend to rotate at the same speed.

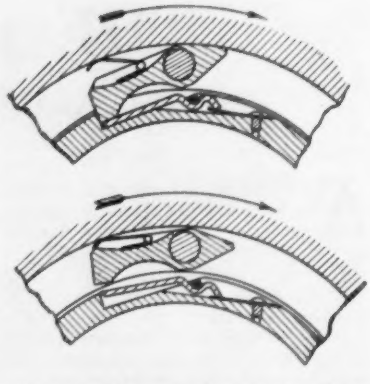
As can be seen from the illustration, the weights are rollers mounted on angle-section links pivoted between lugs at the periphery of a circular carrier. Under the influence of centrifugal force, the weights swing forwards and outwards and roll radially over the cam profile to force forward the assembly comprising the cam ring, reaction plate and presser plate.

The carrier is splined on to a sleeve, the rear end of which projects through a hole in the centre of the clutch cover. A disc is mounted on this end of the sleeve. The front face of the disc is lined with a friction material. Normally, the friction facing bears against the clutch cover and transmits the drive through this sleeve to the centrifugal-weight assembly. The whole assembly on the sleeve is carried by a ball bearing; the outer race is housed in the clutch cover and the inner race is mounted on a boss round the centre of the centrifugal-weight carrier.

The sleeve and disc assembly is spring-loaded to clamp the friction

A clutch unit completely assembled

facing against the rear face of the clutch cover. This loading is effected by compression springs interposed between two circular plates carried round the sleeve, one against the inner race of the ball bearing and the other retained by a circlip in a groove near the forward end of the sleeve. Thus, the whole assembly tends to rotate with the clutch casing so that the centrifugal force on the weights and therefore the torque that can be transmitted through the clutch increases as the engine speed rises. The profile of the cam ring, of course, is designed to give a more progressive engagement than

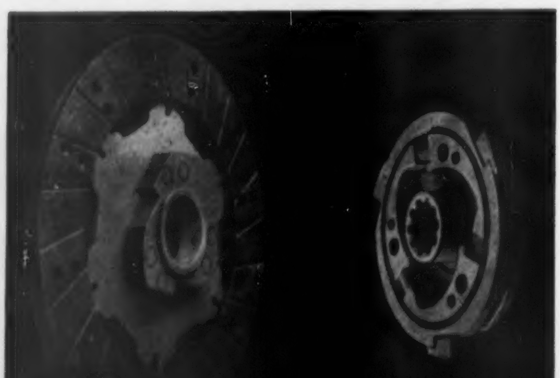


Above, ratchet engaged with the vehicle stationary. Below, ratchet disengaged by centrifugal force

would be possible if the weights were to bear on a flat surface.

Disengagement is effected by an annular electro-magnet bolted to the front end of the gearbox. When the magnet is energized, it attracts the disc on the rear of the sleeve. This disengages the friction drive between the disc and the clutch casing and also brakes the centrifugal-weight carrier assembly to stop it. Thus, the clutch is disengaged in the following manner. As the carrier and weight assembly slows down, the reaction plate moves to the rear until it comes against the stop plates on the lugs of the presser plate. Then, the retraction springs withdraw the presser and reaction plate assembly clear of the driven plate.

From the illustration of the performance characteristics of the clutch, it can be seen that the torque that can be transmitted as the speed decreases is greater at any given speed than that obtainable when the engine is being accelerated after engagement. This hysteresis effect is due to friction in the clutch mechanism. It is an advantage in that it enables the engine to be used as a brake even at speeds approaching idling.



The driven plate and its hub assembly

Operation

With the engine idling, the clutch is always disengaged, because the centrifugal force on the weights is insufficient to overcome the tension in the presser plate withdrawal springs. Therefore, to pull away from rest, the sequence of events is as follows. First, the appropriate gear is selected. Then the throttle is eased open, and the engine speed, and therefore the centrifugal force on the bob weights, is gradually increased until the drive is fully taken up.

To change gear, either up or down, while the vehicle is in motion, it is only necessary to move the gear shift lever. During the initial movement of this lever, the interrupter switch, which is incorporated in it, closes the circuit to the electro-magnet. This withdraws the drive-sleeve of the centrifugal-weight assembly rearwards and slows it down until it stops. As the centrifugal force on the weights is reduced, the presser plate is withdrawn by its return springs and the clutch is disengaged. Synchronization is effected by the synchro cones in the gearbox as the gear change is completed. When the driver releases the lever, the circuit to the electro-magnet is broken, and the drive sleeve of the centrifugal-weight assembly moves forward again under the influence of its return springs until its friction facing is clamped against the clutch cover. Thus, the weights rotate again and the drive is taken up by the centrifugal clutch. Doubtless, slight hysteresis would only give a more gentle take-up.

THE I.B.C.A.M. COMPETITIONS

Conditions for the Drawing and Handicraft Sections

PARTICULARS are now available in connection with the drawing and handicraft competitions arranged by the Institute of British Carriage and Automobile Manufacturers, in association with the Society of Motor Manufacturers and Traders, the National Federation of Vehicle Traders, and the Worshipful Company of Coachmakers and Coach Harness Makers of London.

Of the five drawing competitions, the first calls for a coloured drawing of a four-door six-seater saloon on a wheelbase of 9 ft, together with separate sketches, in line or colour, of details such as interior trimming, luggage boot, instrument board and special features, suitable for use for a sales campaign or catalogue illustration. The coloured drawing to be to an approximate scale of ¾ in to the foot, in three-quarter front view perspective, hand brushed or finished by mechanical means. This competition is open to all persons of British nationality, without age limit.

Competition No. 2 is open to persons of British nationality, without age limit, and is for a general arrangement drawing to a scale of 1½ in to the foot, of a four-door six-seater saloon on a wheelbase of 9 ft, with sufficient details to show method of construction; various views to be shown to include elevation, half plan, half front and half rear. Elevation to show car travelling from right to left, and half views to be of left hand (near) side only.

Competition No. 3 is for an outline drawing of a four-door, six-seater saloon on a 9 ft. wheelbase; drawing to be to a scale of 1½ in to the foot; seating arrangements to be indicated. Various views to be shown are to include elevation, half-plan, half front and half rear. Elevation to show car travelling from right to left, and half views to be left hand (near) side only. This competition is open to students of any technical school or employees in British workshops under 21 years of age and to competitors under the age of 23 at closing date for receipt of entries who have served not less than 12 months in H.M. Forces, but the competition will not be sub-divided into separate sections for the two age groups.

Competition No. 4 is open to persons of British nationality, without age limit, and calls for a general arrangement drawing to a scale of ¾ in to the foot, of a single-deck stage carriage of metal construction mounted on an underfloor engined chassis. Coachbuilder's chassis drawing will be supplied. Overall dimensions to be 30 ft long by 8 ft wide, vehicle to comply in all respects with current Ministry of Transport regulations appertaining to single deck vehicles. Side elevation, front, rear and plan views to be shown to the same scale, with half cross-section giving details of construction to a scale not exceeding 1½ in to the foot. The body envisaged must be capable of carrying in reasonable comfort, the maximum possible number of seated and standing passengers during peak periods, and be capable of seating not less than 24 passengers at any time. Separate entrance and exit doors to be provided, with general layout suitable for one-man operation during non-peak periods. Body weight and maximum front and rear axle loads in both laden and unladen condition to be specified, with clear indication of method of arriving at calculated figures.

Competition No. 5, open to persons of British nationality, without age limit, requires a general arrangement of an accident ambulance, to a scale of 1½ in to the foot; to comply in all respects with current Ministry of Transport regulations. Interior equipment to be designed to accommodate two stretcher patients, or one stretcher patient and four to six seated patients. Constructional details to be given, with full size sections through various parts of the body; body mounting details also to be shown. Chassis drawing will be supplied to competitors.

The Handicraft competition is open to students of any technical school, or employees in British workshops under the age of 21; it is also open to competitors under the age of 23 at closing date for receipt of entries who have served not less than 12 months in H.M. Forces, but competition will not be sub-divided into separate sections for the two age groups. Competitors may submit entries of handicraft in any branch of the industry.

In the woodwork sections the use of machinery, other than for the rough sawing of timber, is not permitted. Blue prints will not be issued, and it is open to competitors to submit any specimen of their handicraft they may select. Very large or very small specimens should be avoided.

All entries, whether drawings or specimens of handicraft work, must be delivered securely packed to the Institute of British Carriage and Automobile Manufacturers, 50 Pall Mall, London, S.W.1, on or before Saturday, May 14th, 1955.

PNEUMATIC SUSPENSION SPRINGS

Some of the Fundamental Considerations

SINCE the purpose of vehicle suspensions is to ensure a comfortable ride, a design, to be satisfactory, must meet certain requirements regarding vibration characteristics with particular reference to their effect upon the passengers. A considerable amount of research has been devoted to this matter[1] and the main results are summarized in Fig. 1. According to Reiher and Meister, passengers should not be subjected to vibrations in excess of those of region 3 on the curves, whilst Helberg and Sperling consider that the limit is set by a ride factor of 2·5. The data obtained by the two last - named investigators form the basis for ride evaluation of German railway rolling stock, the relevant limits being shown in Fig. 1. Janeway concludes that for automobiles the comfort limit for comfortable ride within a frequency range of 1-6 cycles/sec is given by $af^3=2$, the limiting value of the jerk (d^3s/dt^3) being 40 ft/sec^3. This value appears somewhat on the high side, and an analysis of Fig. 1 suggests that within the frequency range of 1-3 cycles/sec, a more desirable value will be given by $af^3=1$. In any case, to ensure the same degree of comfort it will be necessary to reduce the amplitude of vibration as the frequency is increased.

The degree of comfort of the passenger depends upon the natural frequency of the suspension, the frequency of the disturbances imposed by road irregularities and the amplitude of the body displacement. To obtain a clear picture of the effect of some of the factors involved, it is necessary to make certain simplifying assumptions and to ignore the effect, upon vibration characteristics, of seat cushions, tyres, vehicle mass distribution, different front and rear spring characteristics and position of the centre of gravity. The problem will be considered on the basis of a single degree of freedom system. For springs having a linear characteristic, that is, with which the deflection s, in, is proportional to the load P, lb, the natural frequency f_n, cycles/sec, is given by:

$$f_n=3 14\sqrt{1/s}$$

whilst the transmissibility or magnification factor e=amplitude of forced vibration/amplitude of disturbance, which equals transmitted force/impressed force, is obtained from:

$$e=\frac{1}{1-(f/f_n)^2}$$

where f is the frequency of the disturbance.

It will be noted that the value of f, as well as of the impressed force, has a considerable influence on the suspension performance. Unfortunately, the available data are limited. Generally, over rough roads, the value of f varies between 800 and 1900 per mile, with 1000 per mile as a fair average.

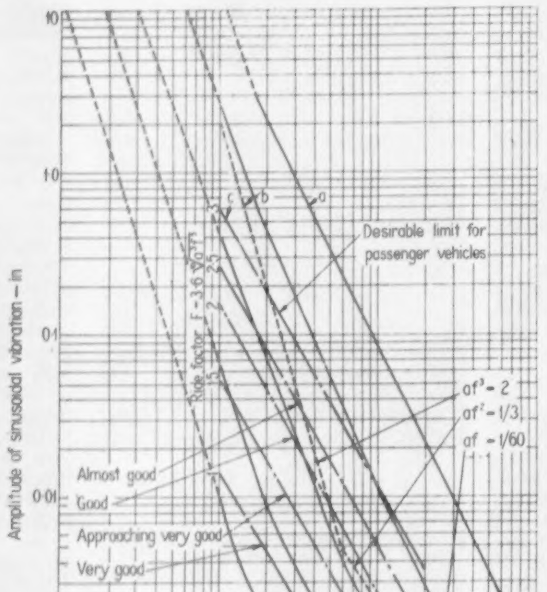


Fig. 1. The effect of vibration on passengers

The ratio of spring deflection at no load to that at full load can vary within wide limits, even with passenger vehicles. For omnibus trailers, it amounts to as much as 1 to 3. This, in turn, means that the ratios of f_n can vary within a range of 1 to $\sqrt{3}$; therefore it may not be possible to ensure satisfactory riding qualities over the entire load range. In any case, it is desirable to maintain a constant natural frequency irrespective of the load imposed upon the suspension, that is, the suspension should have a non-linear characteristic, the stiffness c, lb/in, increasing with the load P, lb. This type of characteristic is sometimes referred to as ***supra-linear.***

The stiffness of a spring with a non-linear characteristic is determined in accordance with a differential equation referring to a point of the characteristic $c=dP/ds$, where s, in, is the spring deflection. From Fig. 2, the value of c is determined by $\tan\beta$. Since the circular frequency of the vibration is $\omega_n=\sqrt{c/m}$, sec^{-1}, the requirement f_n=constant means that ω_n, and with it the value of c/m, where $m=W/g$, lb-sec^2/in, and W=the spring-borne weight, lb, must remain constant over the entire range of loads. With $W=P$:

$$c/m=(dP/ds)/(P/g)$$
$$=\omega_n^2,\ \text{sec}^{-2}$$

or,

$$P=(g/\omega_n^2)(dP/ds)$$
$$=C\,dP/ds$$

where $C=g/\omega_n^2$ is a constant determined by the required value of ω_n. The solution of this equation is:

$$\int ds=s=C\int dP/P$$
$$=C(\log_e P-\log_e P_o)$$

Consequently,

$$P=P_oe^{s\omega_n^2/g}$$

where $e=2{\cdot}718$, and P_o is an arithmetical value required to plot the load versus deflection curve for a constant value of ω_n or f_n. Generally, it is advisable to determine the spring characteristic on the basis of $P_m=3P_o$, where P_m is the minimum load carried by the spring[2].

A number of curves for constant values of f_n or ω_n are plotted in Fig. 4. It should be noted that below the value $P/P_o=e$, the theoretical values shown by dotted lines deviate from what is practically possible, the actual curves starting from the point of origin and not at $P/P_o=1$. It is possible with a number of different spring arrangements to obtain a close approximation to the requirement f_n=constant. For example, this can be done with wishbone suspension, torsion bars, and suitably shaped rubber buffers: the

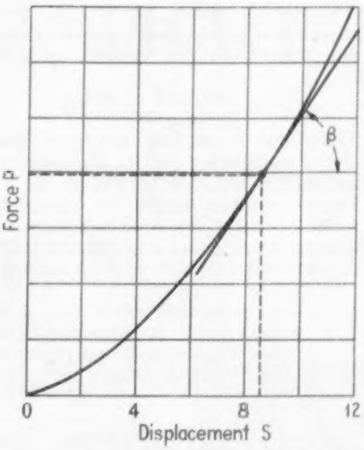


Fig. 2. Non-linear spring characteristic

resultant characteristic is plotted in Fig. 3, in which the no load condition is $P_m/P_o=3$.

The design of a controllable pneumatic suspension of improved riding qualities calls for a performance analysis with particular reference to the load-deflection characteristics as well as the heat dissipation aspects of the units. To simplify the basic considerations, the suspension is assumed to consist of a piston moving in an airtight cylinder, and that no air leaks past the piston. The pressure in the cylinder is given by: $p_o=W/A$, lb/in^2 gauge, where A, in^2, is the piston area. The force resisting additional deflection depends only upon the volume of air compressed by the piston, that is, $p=p_o(V_o/V)^\gamma$, so that:

$$\begin{aligned} P &= pA = p_o A(V_o/V)^\gamma \\ &= W[V_o/(V_o-sA)]^\gamma \\ &= W[1/(1-s/H_o)]^\gamma \end{aligned}$$

where H_o, in, is the working length of the air column under the static load P_o. For air undergoing adiabatic compression, $\gamma=c_p/c_v=1{\cdot}4$. The resultant characteristics, for a number of H_o values and an initial static deflection of 4 in, are plotted in Fig. 4. It should be noted that the values of s become negative as soon as $s<4$ in. As shown in Fig. 4, it is possible to maintain a constant natural frequency within a range of load ratios of about 2 to 1. Outside this range the desired frequency can be maintained by changing the value of H_o by altering the air pressure, a practice adopted with some recent omnibus suspensions.

Transmissibility curves of air springs are generally as shown in Fig. 5. The curves for the lower range of loads are rounded off at the peak, this represents the effect of damping in the system. As the impressed frequency increases with speed, or due to road conditions, the value of transmissibility will suddenly increase at f_y; the shaded hysteresis loop is due to system instability in the range concerned[3].

The thermal aspects of air springs will now be considered for a four-wheel omnibus in which each of the eight spring-cylinders supports a load of 1,400 lb when the bus is empty, and 2,800 lb fully laden. With a static deflection of 4 in and a corresponding effective spring length $H_o=10$ in, a constant natural frequency of about 90 cycles/min can be ensured over the working range of the spring. When the omnibus is fully laden, the deflection will increase from 4 in to 7·85 in, Fig. 4, whilst a dynamic load equivalent of 0·25 of the static value will further increase this to 8·35 in. It should be noted that the non-linear characteristic reduces the deflection due to the shock load to about 0·5 in, as compared with 1 in for a normal spring.

If the cylinders are filled at atmospheric pressure and then compressed by the load of 1,400 lb, the air pressure will increase to 5·88 lb/in^2, assuming isothermal compression when reducing the length of the air column from 14 in to 10 in. Since the piston diameter of 17·4 in needed under these conditions is unduly large, the air must be compressed initially. If the piston diameter is limited to 6 in, the air pressure required will be 35·3 lb/in^2; this increases to 49·5 and 99 lb/in^2 under loads of 1,400 and 2,800 lb respectively.

The work done by the springs will be converted to heat, and the effect of the main parameters will now be considered for the omnibus running at 30 m.p.h. over a rough road in an ambient temperature of 100 deg F. If the number of oscillations is 1,000 per mile and the dynamic spring deflection 0·5 in, the mean piston velocity will be

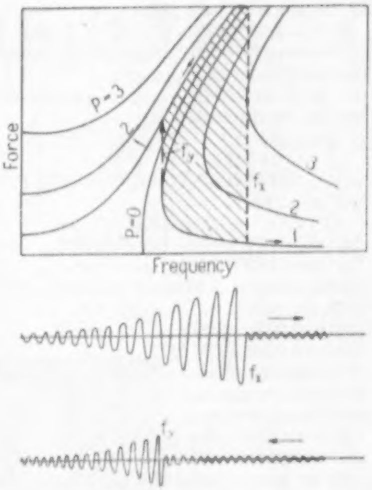


Fig. 5. Resonance characteristics of non-linear springs

0·417 ft/sec. The compression will follow a polytrope, the exponent of which is assumed as $m=1{\cdot}333$. The pressure p_2 at the end of the compression stroke can be obtained from:

$$\begin{aligned} p_2/p_1 &= (V_1/V_2)^m \\ &= (141{\cdot}5/127)^{1{\cdot}333} \\ &= 1{\cdot}156 \end{aligned}$$

where V is the volume, in^3. If $p_1=99$ lb/in^2, $p_2=114{\cdot}5$ lb/in^2, the temperature t_2 at the end of the compression stroke is given, for $t_1=100$ deg F or $T_1=560$ deg F absolute, by:

$$\begin{aligned} T_2/T_1 &= (V_1/V_2)^{m-1} \\ &= 1{\cdot}156^{0{\cdot}333} \\ &= 1{\cdot}035 \end{aligned}$$

so that $T_2=580$ deg F absolute and $t_2=120$ deg F.

For air at 100 deg F, $\gamma=c_p/c_v=1{\cdot}398$ and $c_v=0{\cdot}172$ B.Th.U/lb-deg F, so the specific heat of the polytrope will be:

$$\begin{aligned} c_m &= c_v(m-\gamma)/(m-1) \\ &= 0{\cdot}172(1{\cdot}333-1{\cdot}398)/(1{\cdot}333-1) \\ &= -0{\cdot}0335 \end{aligned}$$

The work done amounts to:

$$\begin{aligned} L &= (R/m-1)(t_1-t_2) \\ &= (53{\cdot}3/0{\cdot}333)(-20) \\ &= -3{,}200 \text{ ft-lb/lb} \end{aligned}$$

or

$$\begin{aligned} L &= 778(c_m-c_v)(t_2-t_1) \\ &= 778\,(-0{\cdot}0335-0{\cdot}172)\,20 \\ &= -3{,}200 \text{ ft-lb/lb} \end{aligned}$$

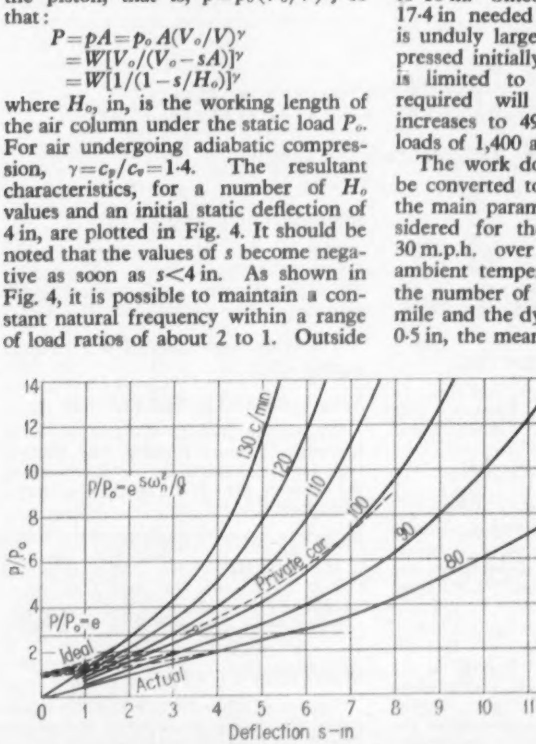


Fig. 3. Constant frequency spring characteristics

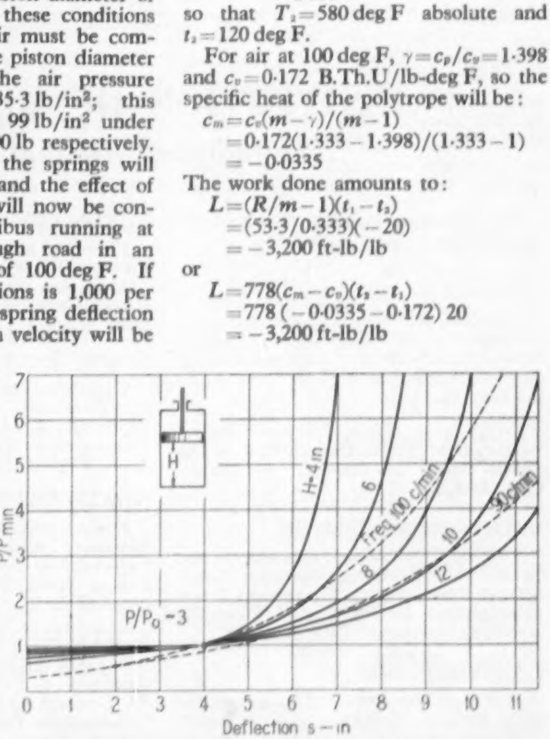


Fig. 4. The effect of pneumatic spring length on characteristic

so, in terms of heat units:

$L' = -3{,}200/778 = 4{\cdot}11$ B.Th.U/lb

The weight of air originally enclosed in the cylinder at a pressure of 35·3 lb/in² is $w = 0{\cdot}039$ lb, so the amount of heat to be dissipated is:

$$Q = c_m(t_2 - t_1)$$
$$= -0{\cdot}0335 \times 20$$
$$= -0{\cdot}67 \text{ B.Th.U/lb}$$

and as a check,

$$Q = L'(\gamma - m)/(\gamma - 1)$$
$$= -4{\cdot}11(1{\cdot}398 - 1{\cdot}333)/0{\cdot}398$$
$$= -0{\cdot}67 \text{ B.Th.U/lb}$$

so the amount dissipated per stroke is:

$$Q_s = wQ$$
$$= 0{\cdot}039(-0{\cdot}67)$$
$$= 0{\cdot}0261 \text{ B.Th.U.}$$

The amount of heat generated per hour at a vehicle speed of 30 m.p.h. and 1,000 oscillations per mile is:

$$Q_{sh} = 0{\cdot}0261 \times 30 \times 1{,}000$$
$$= 783 \text{ B.Th.U/hr}$$

The heat dissipation can be determined by the amount of heat transferred from the working charge to the cylinder walls and thence to the surrounding air. Heat dissipation from the charge to the walls is governed by:

$$Q_s = \alpha A(t_m - t_w)\Delta\theta \text{ B.Th.U.}$$

where A, ft², is the effective cylinder area (in calculating this, the area of both ends will be ignored because of the installation requirements), t_m, deg F, the mean cycle temperature, $\Delta\theta$, hr, time per cycle, whilst α, k cal/m^2-deg C-hr, is the heat transfer coefficient.

According to Nusselt[4],

$$\alpha = 0{\cdot}99\sqrt[3]{p_m{}^2 T_m}\,(1 + 1{\cdot}24\, v_m)$$

where p_m is the mean cycle pressure, T_m the mean cycle temperature and v_m the mean piston velocity, all in metric units. Here, $p_m = 8{\cdot}27$ atm absolute, $T_m = 317$ deg K, and $v_m = 0{\cdot}127$ m/sec, so

$$\alpha = 0{\cdot}99\sqrt[3]{8{\cdot}27^2 \times 317}\,(1 \times 1{\cdot}24 + 0{\cdot}127)$$
$$= 31{\cdot}9 \text{ k cal}/m^2\text{-deg C-hr}$$

or,

$$\alpha = 31{\cdot}9/4{\cdot}88$$
$$= 6{\cdot}53 \text{ B.Th.U/ft}^2\text{-deg F-hr}$$

At a length of 5·65 in, the effective cylinder wall area is 0·735 ft², so,

$$Q_s = 6{\cdot}53 \times 0{\cdot}735 \times (t_m - t_w) \times 0{\cdot}0000333$$
$$= 0{\cdot}00016(t_m - t_w)$$

and

$$t_m - t_w = 0{\cdot}0261/0{\cdot}00016$$
$$= 163 \text{ deg F}$$

Consequently to dissipate the amount of heat concerned, the mean temperature inside the cylinder must be some 63 deg F above the ambient.

The heat dissipation from a single tube to air, if the direction of the air flow is normal to the tube, can be calculated[5] from:

$$\frac{hD}{k} = 0{\cdot}385\left(\frac{c_p}{k}\right)^{0\cdot3}\left(\frac{DV\rho}{\mu}\right)^{0\cdot56}$$

where h = heat transfer coefficient, B.Th.U/ft²-deg F-hr
k = thermal conductivity, B.Th.U/ft²-hr-deg F/ft
c_p = specific heat at constant pressure, B.Th.U/lb-deg F
μ = absolute viscosity, lb/hr-ft
ρ = density, lb/ft³
D = diameter, ft
V = air velocity, ft/hr

At a vehicle speed of 30 m.p.h., the velocity of air past the spring-cylinders is assumed to be about 10 m.p.h., so $h = 3{\cdot}57$ B.Th.U/ft²-deg F-hr. The cylinder area exposed to the cooling air amounts to about 1·85 ft², so the required temperature difference between the cylinder and the ambient air will be:

$$T.D. = 783/3{\cdot}57 \times 1{\cdot}85$$
$$= 120 \text{ deg F}$$

Thus, the controlling factor is the ability to dissipate heat to the ambient air, so the provision of fins or use of larger cylinders might be advisable when operating under arduous conditions or if the spring units are not fully exposed to the air flow past the vehicle.

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INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during March:—

LONDON

Tuesday, 8th March, 5.30 p.m., at 1, Birdcage Walk, Westminster, S.W.1. Paper: "Analysis and Interpretation of Service Records," by A. T. Wilford, B.Sc., C.I.Mech.E.

Friday, 11th March, 5.30 p.m. Paper: "Heavy-duty Truck Development in the United States of America," by Robert Cass, M.S.A.E.

BIRMINGHAM CENTRE

Thursday, 17th March, 6.45 p.m., in the James Watt Memorial Institute, Great Charles Street. Paper: "Heavy-duty Truck Development in the United States of America," by Robert Cass, M.S.A.E.

DERBY CENTRE

Monday, 28th March, 7.0 p.m., in the Rolls-Royce Welfare Hall. Paper: "Nuclear Reactors and Power Production."

LUTON CENTRE

Thursday, 24th March, 7.30 p.m., in the Assembly Room, Luton Town Hall. Paper: "Heavy-duty Truck Development in the United States of America," by Robert Cass, M.S.A.E.

NORTH-EASTERN CENTRE

Wednesday, 16th March, 7.15 p.m., in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Analysis and Interpretation of Service Records," by A. T. Wilford, B.Sc., C.I.Mech.E.

NORTH-WESTERN CENTRE

Friday, 18th March, 7.15 p.m., in the Engineers' Club, Manchester. Paper: "Heavy-duty Truck Development in the United States of America," by Robert Cass, M.S.A.E.

SCOTTISH CENTRE

Monday, 21st March, 7.30 p.m., in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Heavy-duty Truck Development in the United States of America," by Robert Cass, M.S.A.E.

WESTERN CENTRE

Monday, 14th March, 6.45 p.m., in the Royal Hotel, Bristol. Paper: "Heavy-duty Truck Development in the United States of America," by Robert Cass, M.S.A.E.

The following meetings will be held during April:—

NORTH-EASTERN CENTRE

Wednesday, 20th April, 7.15 p.m., in the Chemistry Lecture Theatre, The University, Leeds. Address by the Chairman of the Centre, Mr. J. L. Hepworth, B.Sc., M.I.Mech.E.

NORTH-WESTERN CENTRE

Monday, 18th April, 7.15 p.m., in the Offices of Messrs. Leyland Ltd., Manchester. Paper: "Analysis and Interpretation of Service Records," by A. T. Wilford, B.Sc., C.I.Mech.E.

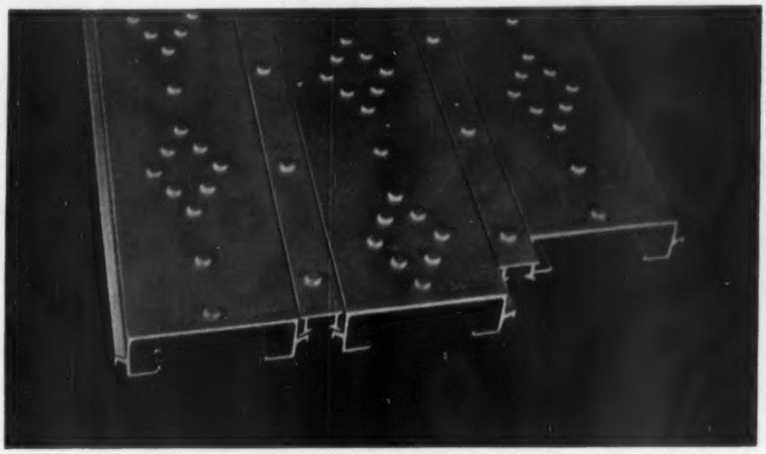
WESTERN CENTRE

Thursday, 28th April, 6.45 p.m., in the Royal Hotel, Bristol. Paper: "Matching a Diesel to Light Road Vehicles," by M. Vulliamy, M.A., A.M.I.Mech.E.

LIGHT ALLOY COMMERCIAL BODYWORK

Birmetals System Based on Standardized Extruded Sections

IN all classes of commercial vehicles a worthwhile reduction of weight can be secured by the use of light alloy bodywork. The aluminium alloys used are approximately only one-third the weight of steel, but owing to their lower modulus of elasticity, however, load-carrying and stress-bearing members must be of correspondingly greater section thickness or have a relatively improved section modulus. The best use of the material can be secured only by careful design of the structure generally and of the individual sections employed. These sections may well differ considerably from the conventional forms established for steel; to secure the most economical disposition of the material they are produced by the extrusion process. A net saving in weight of approximately 50 to 60 per cent can usually be realized.



Assembled anti-slip, heavy-duty flooring (Patent applied for)

The first cost of a light alloy body is higher than an equivalent steel structure and the advantages are obtained in operation. A lighter body enables a higher pay-load to be carried while retaining the same gross weight rating. Fuel consumption will be lowered and tyre wear lessened when the vehicle is running light. Should the same maximum load be carried these economies will apply during all running, whether fully loaded or empty. In some instances, while carrying the same load, the reduction in gross weight is sufficient to bring a 20 m.p.h. vehicle into the class that is legally operated at 30 m.p.h., with consequent increase of earning capacity. An incidental advantage, to some extent offsetting the higher first cost, is the high scrap value of the light alloy body when the vehicle is written off at the end of its useful working life.

A range of specially developed extruded sections is produced by Birmetals Ltd., Quinton, Birmingham, in BB016 WP heat-treated magnesium silicide alloy for all classes of commercial vehicle bodywork. This alloy has the following mechanical properties:

0·1 per cent proof stress	14-19 ton/in^2
Ultimate stress	17-23 ton/in^2
Elongation	10-25 per cent

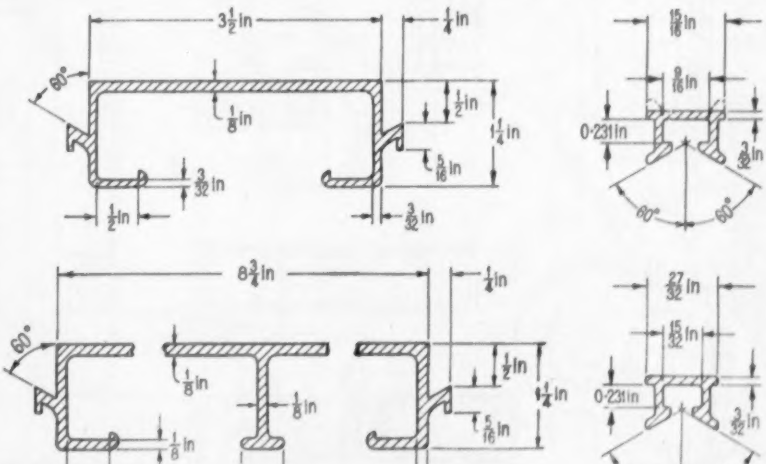
The sections are so designed that they are adaptable for various parts of the body structure, thereby minimizing the number of different sections to be carried by the body builder. It will be noted that hollow sections, more expensive to produce, have been avoided and all extrusions are of the "open" type. An outstanding feature is a new type of interlocked flooring.

Underframe

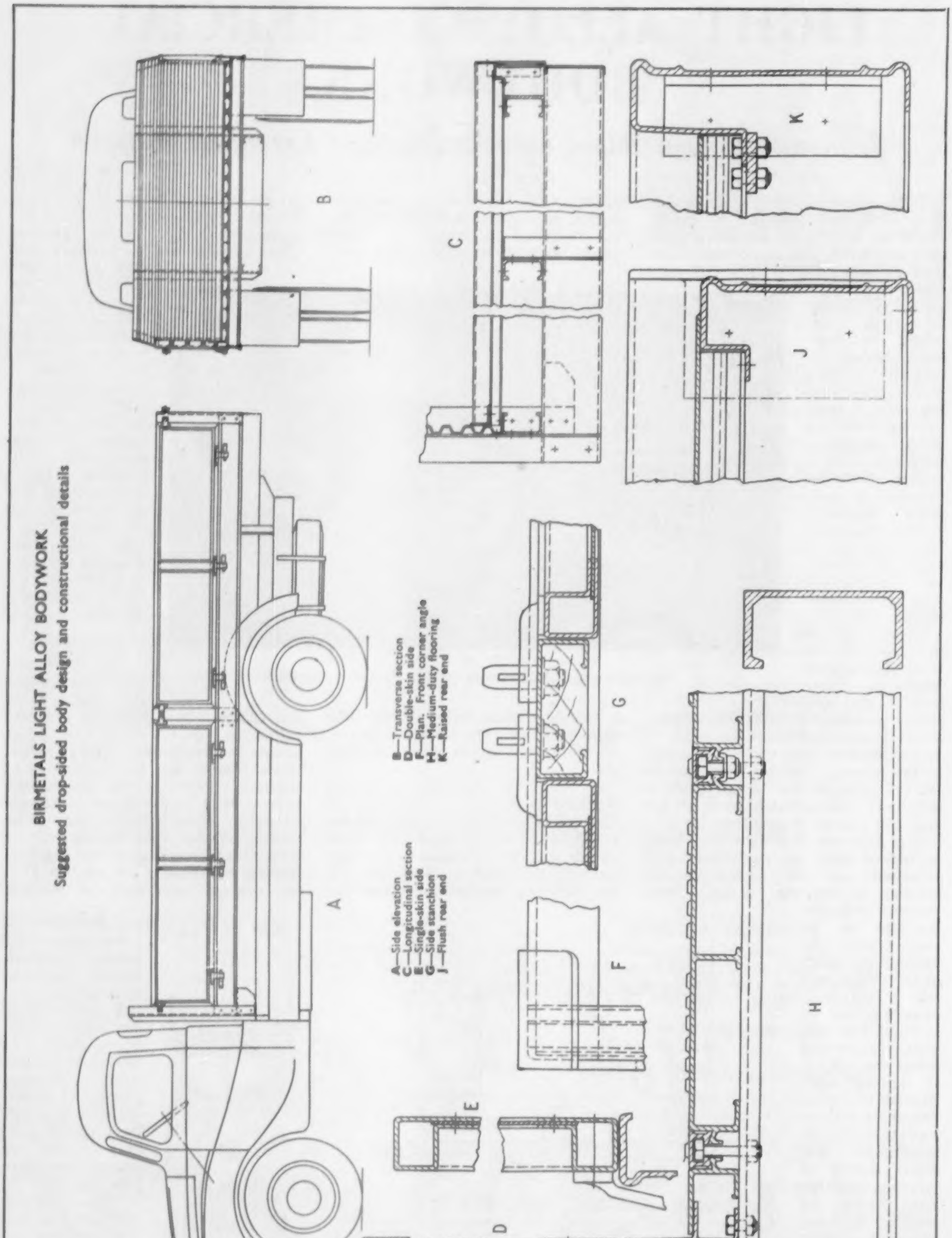
The longitudinal bearer support channels are 6 in × 2 in × $\frac{5}{16}$ in flanges for platform bodies or up to 9 in × 3½ in × ½ in flanges for heavy-duty tipping bodies. Cross bearers are 4 in × 2 in × $\frac{5}{16}$ in channels, bolted to the longitudinal bearers with 2½ in × 2½ in × ¼ in angle stabilizers at each intersection. The spacing of the cross bearers will be determined according to the platform loading but should not exceed 27 in for average conditions of service.

Flooring

For heavy-duty vehicles the plank section is 3½ in wide × 1¼ in deep, and comprises a box channel with an angled projecting rib on each side. Adjoining planks are locked together by a capping section suitably angled to engage the plank ribs and provide a flush surface over the junction. The capping accommodates the hexagon head of a standard $\frac{5}{16}$ in diameter B.S.F. high-tensile steel bolt, free to slide but not to turn,



Heavy- and medium-duty planking sections and $\frac{5}{16}$ in and ¼ in bolt size lock strip sections



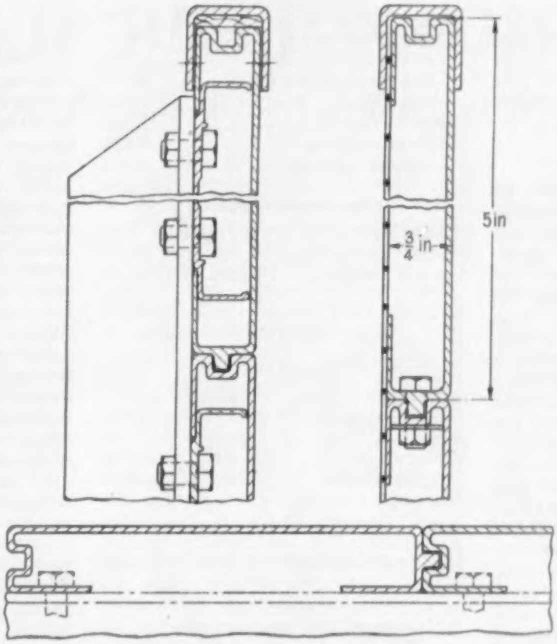
BIRMETALS LIGHT ALLOY BODYWORK

Suggested drop-sided body design and constructional details

A—Side elevation
C—Longitudinal section
E—Single-skin side
G—Side stanchion
J—Flush rear end

B—Transverse section
D—Double-skin side
F—Plan. Front corner angle
H—Medium-duty flooring
K—Raised rear end

by which it is drawn down to clamp the planks by means of a channelled grip washer engaged on the underside of the plank ribs. These grip washers may be of any appropriate length, being merely cut from an extruded section. At the cross bearers, longer bolts are carried through the flange of the bearer and provide location. By thus locking the planks together between the bearer supports an exceptionally rigid floor is obtained, well suited to resist the effects of dumping when loading by grab without bursting at the joints or distorting. A dielectric paint is applied at all mating surfaces and a coating of "Seelastik" will render the joint completely watertight, as may be necessary for such applications as the bodies for malt or ballast trucks. Lock capping and grip washer sections for ¼ in diameter bolt fixing are also available.



Tongued and grooved planking sections (G. C. Smith (Coachworks) patent)

Medium - duty flooring utilizes a plank section of similar design but 8¼ in wide and centrally supported by a rib seating on the cross bearer. The same lock capping, bolts and grip washers are used and the two types of planks can be assembled together. Thus it is possible to build a floor with a specially strengthened area if desired, or to make use of alternative width sections to obtain a specific width of floor. Accommodation in respect of width of floor is also possible by sawing either an 8¼ in or 3½ in plank to provide "one-legged" sections which are fitted at each side rave.

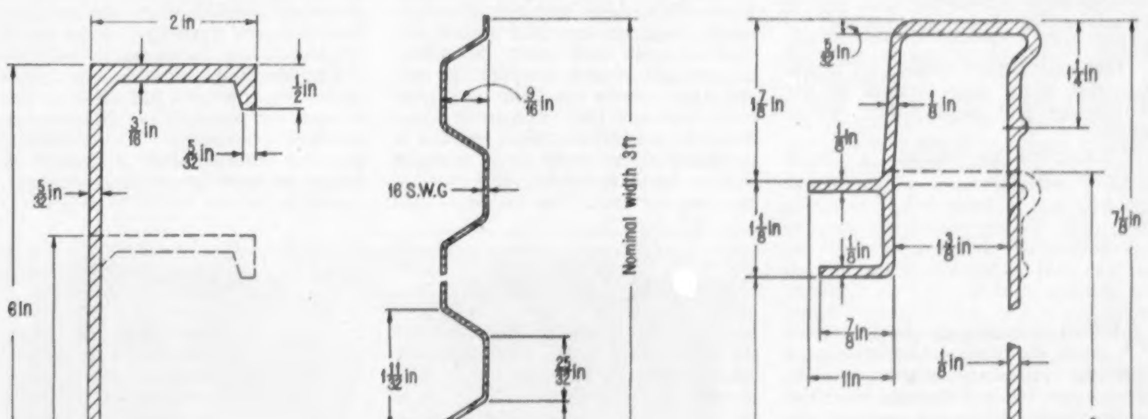
Floor sections are supplied with either a smooth or studded top surface. The studding, which is the subject of a patent application, is formed without adding weight to the section and is arranged in a spaced diamond pattern to afford a non-slip surface in all directions. Where a non-slip surface is required while retaining the facility of sliding boxes, milk churns, drums or similar loads, a special twin-ribbed lock capping section is fitted between the planks. The ribs stand to the same height as the studs and provide a continuous level on which the load items can be moved. For trucks intended for the transport of livestock, anti-skid cross rails of top-hat section are bolted to the non-slip floor.

Two rave sections are available; a flush type and a raised type projecting 1¼ in above floor level. They combine structural and decorative functions, being stiffened by top and bottom bulges to protect the outer lettering surface from abrasion. Extending for the full length of each side of the body, the rave fits closely over the ends of the cross bearers, to which it is bolted at the upper and lower flanges. A rave is also fitted across the rear end, with angle stiffener gussets at each corner.

At the rear of the platform the ends of the planks are supported on a flat strip bolted beneath the intermediate flange of the raised rave section or on the intermediate flange of the flush type section. In both cases the upper flange may be sawn off.

Headboard

For the headboard a Birmabright 16 S.W.G. corrugated sheet is used, with the corrugations running horizontally. Produced under a press brake from an 8 ft by 4 ft sheet, it is toughened by work-hardening and is larger and lighter than the usual extruded corrugated section. An offset boxed capping gives stiffness to the upper edge and at the base an angle finisher is bolted to the floor and the headboard. The main centre supports are two 3 in × 2 in × ¼ in angles extending down to the underframe longitudinal channels to which they are bolted. At each side a 3 in × 2 in × ¼ in



Longitudinal and cross bearer, headboard and rave sections

angle forms a corner finisher. It is bolted to the outer face of the side rave and, extending below it, is gusseted by either a 3 in × 2 in × ¼ in angle or a 4 in × 2 in channel to the lower flange of the rave.

Sides and ends

Since the plank sections can be firmly connected together by means of the lock capping, and the securing bolts are contained within the depth of the section, it is possible to use these same sections to make up double-walled sides and ends. The planks are bolted up with the ¼ in lock capping to form the required area, a light alloy sheet is riveted to the outer side to close the sections and a U-type capping section is fitted all round. On double-skinned hinged sides or tail-boards, lengthened securing bolts are fitted at appropriate points to pass through the outer panel and the hinge bands. Thus no protruding bolt heads are necessary on the inside surface. Whether the 3½ in or the 8¾ in planks, or assemblies of both sizes, are used, this construction is more economical in weight and cost than one involving the use of hollow extrusions of similar thickness.

As an alternative, a single-skinned construction can be employed. This consists of a light alloy panel of suitable gauge framed with an offset boxed capping, mitred at the corners and riveted to the sheet.

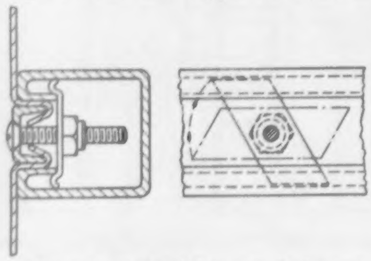
Side stanchions can be made from the 3½ in wide floor plank by cutting off the angled rib on each side. The top run of the side rave is apertured to pass this trimmed section, which is then bolted up with its outer face to the inner vertical face of the rave. Additional security for the anchorage is obtained by an angle gusset to a cross bearer. Alternatively, a special light hollow section, as adopted by Bowyer Bros. (Congleton) Ltd., can be bolted to the outer surfaces of the sides.

The approximate weights, in lb per foot run, of the main sections for the heavy-duty and medium-duty bodies are:—

Underframe longitudinal	2·105
Cross bearer	1·735
3½ in planking	1·005
8¾ in planking	2·118
Lock capping—$\frac{3}{16}$ in	0·264
Lock capping—¼ in	0·253
Rave—flush	1·785
Rave—raised	2·320
Offset capping	0·684
Channel capping	0·474

Lightweight flooring

For lorries of lower capacity and trucks, lighter types of extruded flooring are available. Typical of these is



Lightweight stressed-skin van body construction

the tongued and grooved planking to the patented designs of G. C. Smith (Coachworks). This comprises a boxed channel 5 in × ¾ in which can be closed by a telescoping channel 3¾ in wide. The inner face of the web of the closing channel is ribbed and grooved at each side. Holes for the ¼ in diameter fixing bolts are drilled in the groove and the rib is spaced so that the head of the bolt is locked against rotation. As a flooring it is used in the open form, seating directly on the cross bearers. Similarly, it can serve for sides and ends, closed either by the extruded channel or by a sheet panel and framed by a channel capping.

A new ultra-light extruded flooring is under development. It is specifically designed to replace spruce flooring ¾ in thick without increase in weight but having an improved distributed load capacity and more durability. Lighter and simpler rave sections will also be available for light duty platform lorries. The corrugated sheet floor, to a uniform $\frac{3}{16}$ in depth, is produced in 16- or 18-gauge material in sizes up to 12 ft × 3 ft.

Van bodies

A stressed single-skin van body structure, intended particularly for overseas assembly, uses pre-formed sheet panels over a framing of a special boxed-section channels with re-entrant flanges. The panels are flanged and lipped to overlie the flanges of the frame channels and the edges of adjacent panels are secured by a single channel-section lock strip. This is drawn down by screws and angled spring-steel plate nuts to stress the panel tightly over the framing. In contrast to a conventional lapped and riveted construction, in which the shear value of the panel is limited to the metal from the rivet hole to the edge, this clamping method draws over the entire length of the edge and distributes the shear over the whole width of the panel. The thickness of the panel may, consequently, be reduced to as low as 24 S.W.G. Spring bolting ensures that the panels will not come loose and also permits a slight torsional flexure in operation. The " Hanks " type nut secured to the spring plate is necessarily of the self-locking type, as it is required initially to turn with the screw after insertion in the frame channel to take up its non-rotatable position.

To assemble, the lock strip carrying the screws with the angled plate nuts on their ends is placed in the frame channel, the screws are turned to position the nuts, and the lock strip drawn out to full extent. The panels are then inserted, the lock strip is seated on the panel lips, and the screws tightened to secure. Replacement of a damaged panel, of course, is equally simple.

Panels may be of any convenient size up to a maximum of approximately 10 ft × 4 ft. Such full-length panels are arranged vertically or horizontally and light stiffening or anti-drumming rails may be provided if necessary. The junctions can be satisfactorily made watertight by the use of " Seelastik " at the engaging surfaces.

The drawings show several alternative constructions, but these do not exhaust the possibilities. Many other standard extrusions are available, allowing considerable flexibility in design to meet specific requirements.

CAR LIGHTING

AFTER motoring on the Continent many motorists return convinced that the yellow headlights used in France are far less dazzling than the white lights used in this country. In order to determine whether there are any advantages in using yellow light some new experiments have been carried out at the Road Research Laboratory. A test track, approximately half a mile long, consisting of a straight portion with a curve at each end, was laid out. Man-size rectangles of sacking on wooden frames were set up in two lines, one on either side of the road, and small obstacles were placed in the centre of the track. The dummy men and obstacles were made to reflect about the same amount of light as the average man's suit.

In the test, two cars set off simultaneously from opposite ends of the track. Each car carried two pairs of lamps of the latest French design, and on each car one pair was fitted with yellow bulbs normally used in France, the other pair with clear bulbs. The two pairs of dipped lights on each car were similarly aimed and were adjusted to give the same light output. By tilting the lamps, conditions ranging from slight to severe glare could be arranged in both colours. Twenty drivers took part in the tests. Rather more than half preferred white light when meeting cars similarly equipped to their own. Six drivers thought yellow light slightly less glaring, but all felt the differences between the systems were small. Evidently, changing from white to yellow light will not solve the dazzle problem.

IMPROVED BICERA COMPRESSOR

A New Unit Designed for Commercial Applications

An improved version of the Bicera compressor has been introduced recently by The British Internal Combustion Engine Research Association, 111-112, Buckingham Avenue, Slough, Bucks. The earlier model was designed as an experimental unit rather than for commercial application. That is, ease of instrumentation, adjustment and modification were the prime requirements. The results obtained were so encouraging that a second prototype has been built, tested and developed for commercial use. Like its predecessor, it has advantages of both internal compression and variable displacement at constant speed. Both the units are described and the test results given in two reports by D. W. Tryhorn, B.Sc.(Hons.), which are published by the Association.

Experience gained by testing Roots-type pressure chargers showed that they are far from ideal as a positive displacement type of compressor. When a new design was put forward by a member of the staff it was realized that the disadvantages of the Roots pressure-charging system could be overcome. It was thought that industry would be well served by the construction and testing of a machine working on the positive displacement principle.

The Roots-type compressor

Obviously, the Roots-type compressor could not have withstood the test of time unless it had a number of marked advantages, and it was considered to be most important that further improvements that could be effected in the new type of unit should not be made at the expense of any of these good features. The principal advantages of the Roots-type compressor are its light weight and small bulk. These are due partly to its layout, but much more to the high speed of operation that is possible because of its large port areas. Lubrication is not required in the working space; and reasonable reliability can be obtained, but only when production and fitting limits are carefully controlled. The maximum useful delivery pressure is about 8 lb/in^2, and peak efficiency is somewhere within the range of 2 to 5 lb/in^2; the more efficient compressors usually develop peak efficiency at the lower delivery pressures.

Leakage

The considerable leakage of air past the rotors is a detrimental feature. It not only causes a loss of overall efficiency, but also a rapid drop in delivery pressure as the speed is reduced while the delivery volume per revolution is kept constant. Thus, because of insufficient boost, the engine cannot develop its maximum torque at low speed, that is, when the power is required for acceleration or hill climbing. Production of the rotors is far from easy, since they are complex in shape and the whole of the surface has to be accurate to within fine limits. Moreover, if the rotors were to touch, a severe seizure would be likely to occur. The noise of the Roots-type machine is a further undesirable feature, but in this respect it does not differ greatly from other high-speed air-compressing machines.

Results obtained by the correct matching of the air flow characteristics of the compressor and engine are generally far more important than the peak overall efficiency of the units when operating separately. So far as possible, a pressure-charged engine should receive exactly the quantity of air required to provide good combustion of the fuel at all desired combinations of speed and load. With most current installations, the engine is supplied at all loads with a quantity of air that the pressure charger, designed to suit the maximum load operating requirement condition, displaces.

This results in appreciable wastage of power. For example, when a compression-ignition engine fitted with a Roots-type pressure charger is running at full speed and light load, the pressure charger delivers its maximum output of air, whereas, under these conditions, normal aspiration would be ample. Increasing the compressor efficiency would only make matters worse, since it would increase the compressor delivery pressure and therefore the power wasted.

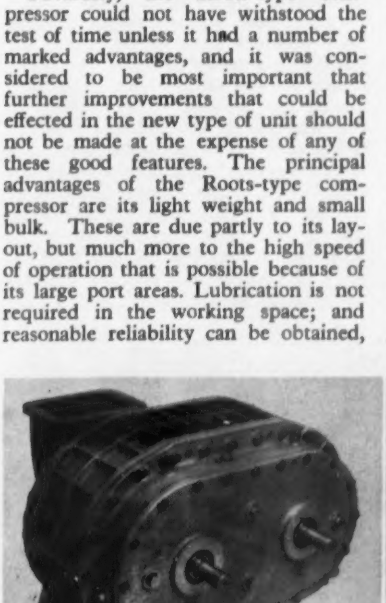
Reduction of the displacement of the pressure charger, on the other hand, would be a great improvement because it would cause a drop in the delivery pressure. When the compressor operates simply as a displacement machine with no pressure rise across it, only the friction and windage losses of the compressor have to be overcome. Therefore, even though the efficiency of the pressure charger, taken on its own, would be zero, the fuel consumption of the engine would be improved. Most engines run at light and medium loads for an appreciable part of their lives, so an air displacement control is likely to give greater fuel economy than a few per cent improvement in the peak overall adiabatic efficiency.

The new Bicera unit

Since a compressor designed on these lines would be well suited to pressure-charging traction-type, compression-ignition engines, it was decided to use aluminium alloy wherever possible in the construction of the new unit. This machine, Fig. 1, with a swept volume of 0·242 ft^3 per revolution, weighs 189 lb; its overall size, including the elbow ducts, is $15\frac{3}{8} \times 20\frac{1}{4} \times 12\frac{3}{4}$ in. The rotor drums are 8 in diameter, and the lobes are $1\frac{1}{4}$ in high × 6 in long. So that the machine could be used as a variable displacement compressor if so required, the inlet valve is fitted with a variable timing sleeve.

This design differs from the first model in a number of ways. For compactness, it is desirable to have as large a rotor length : diameter ratio as practicable. This ratio can be greatly increased if the bearings are at both ends of the rotors, instead of at one end only. In this way the loading on the bearings is reduced, so they can be smaller. As a result it is possible to house the main roller race in the rotor bore, thereby saving space and increasing rigidity.

The outer bearings are built into the inlet and delivery elbow ducts. In practically every application, compressors are coupled to ducting of some kind, so this arrangement does not necessarily take up extra space. In the test machine, the main bearings were oil lubricated and sealed by a piston-ring type oil seal. The outer bearings were grease lubricated. To keep the weight and moment of inertia of the rotors to a minimum, they were built



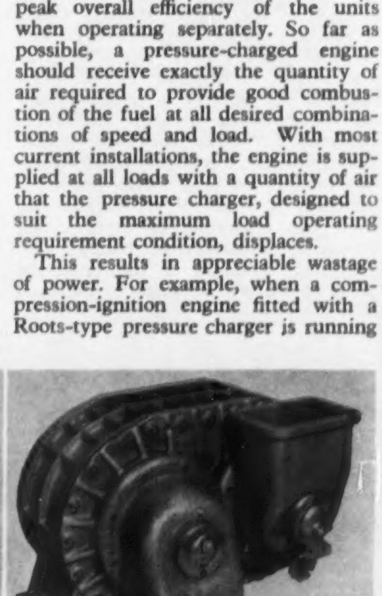


Fig. 1. To keep the weight to a minimum, aluminium is used extensively in the new Bicera compressor

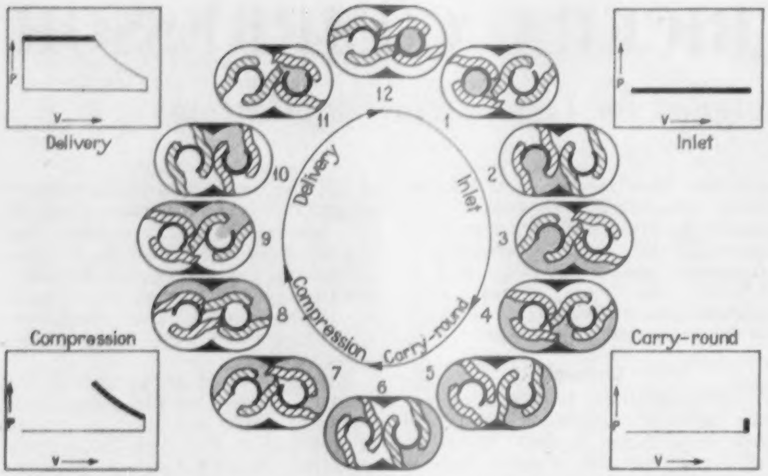


Fig. 3. Cycle of operations of the Bicera compressor. Both flow and internal compression can be controlled during operation by adjusting the cylindrical valves in the rotors

up of aluminium alloy lobe blocks held between a steel disc at the inner end, where the gears are, and a ring at the outer end.

As the lobes pass at the centre of the unit, there is only a small clearance between each and the valve of the opposite rotor, Fig. 2. This gives a good seal, which is not adversely affected by phase tolerance between the rotors. It is a noteworthy feature, since it means that accurate phasing of the rotors, which is essential with the Roots- and screw-types of machine, is unnecessary. The accuracy of phase relationship required is determined by the need for a working clearance between the concave face of the delivery lobe, and the tip of the inlet lobe at the end of the delivery process. Since the period during which the two lobes are passing one another represents only 6 per cent of the cycle, the clearance at this point can be relatively large without seriously affecting efficiency. To improve the air flow path into and out of the working spaces, the port edges are turned at a greater angle from the radial position, as compared with the earlier unit. This has an appreciable effect on the efficiency of the unit.

The cycle

The path of a single charge through one complete cycle is shown pictorially in Fig. 3. Diagrams 1 to 4 show how the inlet air is drawn into the compressor through the left-hand rotor and thus fills the spaces between the lobes. This air is divided into two volumes, known as the *carry round* volumes, and is taken round the peripheries of the rotors until the position shown by diagram 7 is reached; then the lobe of the right-hand rotor dips into the pocket of air carried round by the left-hand rotor, and direct compression begins.

Compression continues until a predetermined pressure, controlled by the position of the tubular delivery valve, is reached. After this, delivery takes place, diagrams 9 to 11. When the position shown in diagram 11 has been reached, the delivery is complete, and the valve closes to stop back-flow. Diagram 12 shows the *dead* period during which the lobes move into position to start the next cycle. The rotor positions 1, 4 and 9 are similar and show the beginning of three cycles during $1\frac{1}{2}$ revolutions of rotation.

Pressure changes in the unit take place as follows. At stage 1, the inlet port is opening; it stays open through stages 2 and 3 until just before stage 4, when it starts to close. Then, the air is carried round the periphery of the rotors without the volume changing, but with the pressure rising as the air leaking past the lobes from the high pressure side is trapped. Only a very small proportion escapes past the second pair of lobes into the inlet space, because the pressure ratio across these lobes is low.

At stage 7, compression begins; it continues through stages 8 to 9, when the valve starts to open. As with piston-type compressors, there is a slight over-compression while the valve is opening initially, after which delivery, at the delivery pressure plus the delivery loss, is effected at a steady rate through stage 10. Just before stage 11, the delivery loss rises slightly and then falls. This is because, at first, the valve starts closing before the rate of delivery has been reduced. Then the rate of delivery falls off rapidly from stages 11 to 12 and is complete about 5 deg before the valve closes.

Design considerations

As mentioned previously, one of the most serious of the losses in compressors in which there is a clearance space between all working parts is that due to leakage. In this unit leakage occurs past the lobes and between the drums. Because of the rolling action of the drums, the clearance between them can be very small, but a reasonable clearance has to be maintained between the lobes and the casing. An advantage of the Bicera compressor is that this leakage is made to perform a first stage of compression. From diagrams 9 to 11, it can be seen that during most of the high pressure part of the cycle the leakage past the lobes enters the carry-round volume of the next cycle, thus raising the pressure. The trapping of this leakage air has three advantages: it not only performs useful work, but is recompressed and delivered on the next cycle, and is prevented from mixing with the inlet air. Should mixing take place, the inlet air would be raised in temperature, and thus the work of compression and the delivery temperature would be increased.

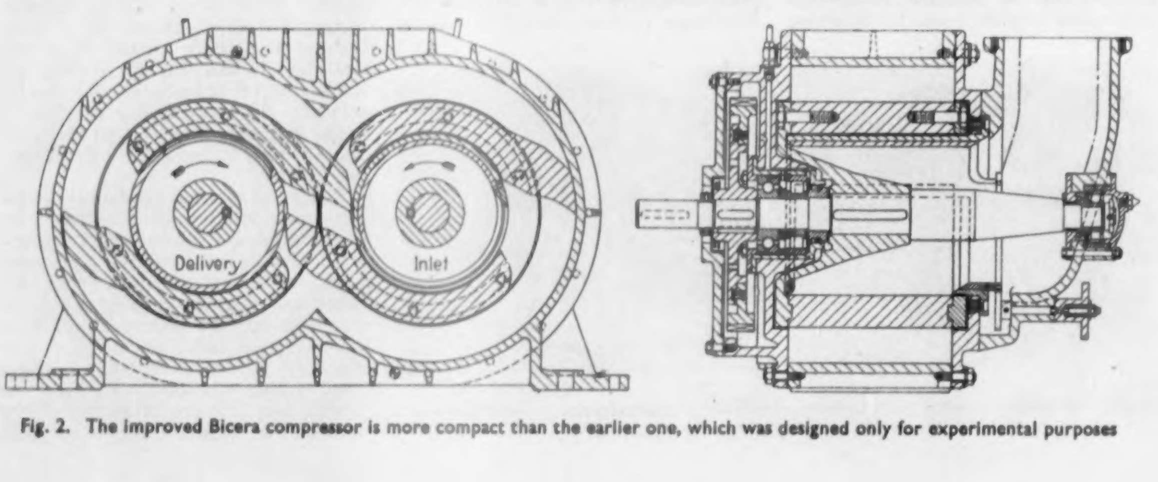


Fig. 2. The improved Bicera compressor is more compact than the earlier one, which was designed only for experimental purposes

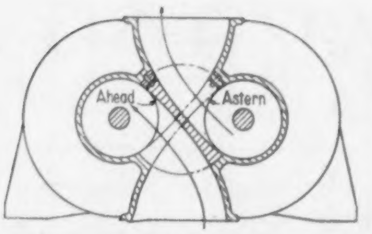


Fig. 4. A vane-type reversing valve can be fitted in the end casing

The volume displaced per revolution per unit length of rotor is approximately the same as that of a Roots-type compressor, so speeds at least as high as those of that type of machine must be contemplated. This means that, if the air inlet and delivery losses are to be kept within reasonable limits, very large ports have to be incorporated, otherwise the whole of the advantage of the improved cycle is liable to be lost. One noteworthy feature of the design of the Bicera compressor is that it allows ample port areas to be incorporated. Another is that compression is effected internally.

For any one set of output conditions, the bulk of the compressor would be comparable with that of a Roots-type unit with a rotor of the same length, but long rotors of small diameter are not practicable. A larger machine would be needed if the maximum displacement per revolution were required at lower speeds. The fact that the air passes through ducts in the end casing may be advantageous in some instances, because these ducts may be arranged to suit the application. In scavenge blowers for reversible two-stroke engines, the reversing valve can be arranged within the blower with practically no increase in size as compared with the single direction machine, Fig. 4.

Inlet pressure loss is more serious than delivery loss because it determines the quantity of working fluid handled: the greater the quantity of air passing, for a constant rate of loss, the greater the efficiency. Therefore, a good feature of this compressor is that the air enters through the hub of the inlet rotor: thus the machine is charged to the extent of the centrifugal head developed by the outward flow. The Roots-type compressor is poor in this respect, since the inlet air enters against the centrifugal head.

An advantage of this type of compressor is that it is possible to alter the timing of the rotor ports while the compressor is in operation. This can be effected by fitting a sleeve in each of the tubular valves. This sleeve can be rotated to give any desired degree of opening. If fitted on the delivery side, it would control the amount of internal compression before delivery and, if it were in the inlet rotor it could be used to control the flow through the compressor. Thus, it is possible to adjust the delivery characteristics of the compressor to suit the requirements of the machine it is supplying.

In theory, the effect of altering the length of the inlet period is shown in Fig. 5. Diagram 1 is for the compressor working with full inlet volume and delivering at the maximum pressure. As the inlet valve is closed the diagram changes progressively to 2 and then to 3. This shows that if the engine requires a constant volume of air from the compressor, the delivery pressure can be made to fall by closing the inlet valve, while the diagram remains almost ideal in shape without any control of the delivery valve. In diagram 3, the inlet and delivery volumes are equal and, with the internal expansion equal to the internal compression, the delivery pressure is zero.

The advantages of the Bicera compressor over the Roots-type can be summarized as follows:

(1) Internal compression by change in volume of the working space.
(2) Degree of internal compression variable.
(3) Quantity of air aspirated per cycle variable.
(4) Reduced leakage losses for a given clearance.

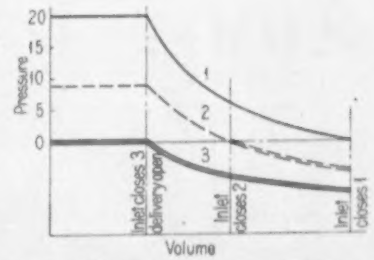


Fig. 5. P.V. diagrams, showing the effect of change of inlet period

(5) Safe clearance between rotors reduced.
(6) Reduced sensitivity of rotor clearance to errors in rotor phasing.
(7) Simpler rotor profile.
(8) Utilization of centrifugal force to assist charging.

Application to a petrol engine

The Bicera variable displacement compressor makes possible the economic pressure charging, by means of a positive displacement compressor, of petrol engines that operate at part load for a considerable portion of their working lives. The Roots-type compressor is designed to pass a constant quantity of air per revolution, so that in this application, the engine will run economically only at full throttle. To satisfy part load requirements, the quantity of air-fuel mixture must be reduced, so a throttle valve is generally fitted. This gives rise to what is known as a large suction loop and provides good engine braking, but is not conducive to fuel economy.

With a variable displacement compressor, it is possible to meter exactly the right quantity of mixture to the engine under all running conditions. That is, the compressor can be made to act as a supercharger at high loads, as a displacement machine without compression at the normal atmospherically-charged engine's full load, and also as an expansion machine at loads below this. When operating as an expansion machine, the compressor delivers

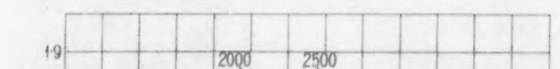


Fig. 6. Relationship between overall adiabatic efficiency, rotor speed, power input, pressure ratio and air flow. Internal volumetric compression ratio 1·05 : 1

Fig. 7. Relationship between overall adiabatic efficiency, rotor speed, power input, pressure ratio and air flow. Internal volumetric compression ratio 1·2 : 1

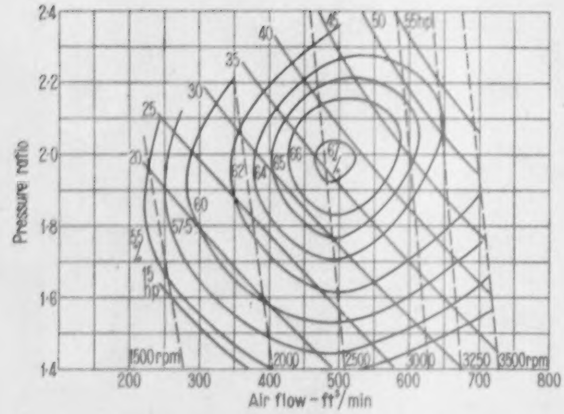


Fig. 8. Relationship between overall adiabatic efficiency, rotor speed, power input, pressure ratio and air flow. Internal volumetric compression ratio 1·4 : 1

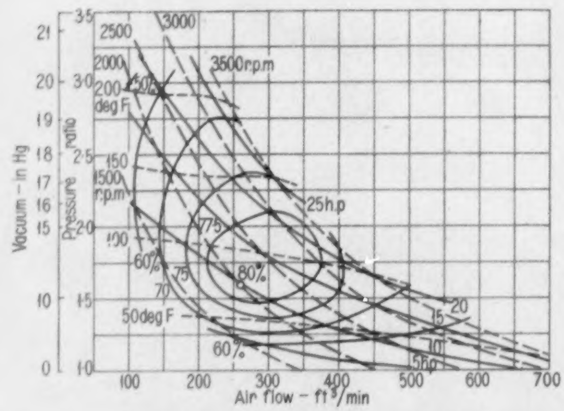


Fig. 9. Relationship between overall adiabatic efficiency, temperature rise, rotor speed, power input, air flow and pressure ratio as an exhauster

power to the engine and so partially makes good the part-load suction loop loss. The fuel consumption of this type of supercharged engine should be better than that of the atmospherically-charged unit. In a vehicle installation, the control from the accelerator pedal would be arranged to operate the supercharger inlet valve control.

Test results

A programme of tests, similar to that carried out on the earlier machine, has been completed. The tests were designed to determine the general performance characteristics over a range of delivery pressures up to 20 lb/in². Figs. 6, 7 and 8 show the lines of constant overall adiabatic efficiency plotted upon the performance curves obtained with three different values of inbuilt compression. The inbuilt compression is regulated by altering the arc subtended by the delivery valve port which, in these three sets of tests, was 120, 90 and 70 deg respectively, to give the internal, volumetric compression ratios of 1·05 : 1, 1·2 : 1 and 1·4 : 1.

The adiabatic efficiencies were computed from the actual power input to the compressor. It has been found that the adiabatic efficiencies based on the temperature rise through the compressor tend to be misleadingly high and are not accurate enough when making evaluations of positive displacement machines. With the Bicera compressor running at high speeds, the temperature adiabatic efficiencies were fairly consistently about 10 to 15 per cent higher than the overall adiabatic efficiencies, but at low speeds the temperature efficiencies sometimes appeared to exceed 100 per cent.

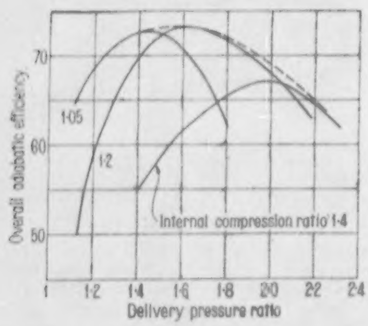


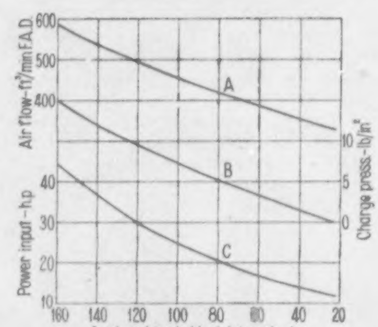
Fig. 10. Relationship between overall adiabatic efficiency and delivery pressure ratio for three values of internal compression ratio. Speed 2,500 r.p.m.

The pressure range over which the inbuilt compression may be held constant can be found from curves of the type illustrated in Fig. 10. These curves show how the overall adiabatic efficiency changes with delivery pressure at a constant working speed. With this unit, the compression ratio of 1·05 would suffice from 0 to 8 lb/in² (gauge), 1·2 from 6 to 15 lb/in² (gauge) and 1·4 from 15 to 20 lb/in² (gauge). In Fig. 10 the broken line represents the maximum efficiency obtainable with an infinitely variable delivery valve, having an inner sleeve similar to the inlet valve. The advantage of having a variable period of inlet valve opening is that the air throughput can be controlled while the machine is running at constant speed. This valve does not have a throttling action; the shortening of the inlet period leads to an adiabatic expansion of the charge, followed by efficient re-compression before delivery.

Fig. 11 shows the results of a test using a variable inlet valve. It was assumed that the compressor was required to run at a constant speed and deliver air to a machine taking, per revolution, a constant volume of air at the delivery pressure. This is similar to the requirement for pressure charging a four-stroke engine. In such a case the maximum charge pressure is required at full load only; at part loads less air is needed and reduction of the parasitic losses of the pressure charger under these conditions would then be worth while. As can be seen from Fig. 11, by progressively closing the inlet valve as the engine load is reduced, the power required to drive the pressure charger can be reduced from 44 to 12 h.p. This should represent a considerable saving in engine fuel consumption.

In many other applications of compressors, some control of throughput is advantageous.

The machine was also tested as an exhauster. These test results, given in Fig. 9, showed it to be equally good for use as an exhauster as for a compressor, but if it were required to run at very low pressures for extended periods, improved grease and oil seals



A—Air flow B—Charge pressure C—Power input

Fig. 11. Performance, using variable inlet valve for the constant delivery condition of 0·123 ft³/rev, at a speed of 3,000 r.p.m.

would be necessary. The high efficiency of operation over the range of 10-20 in of mercury depression and large throughput are an appreciable improvement, as compared with many existing types of machine.

HANDBOOK

THE sixth edition of the Gauge and Tool Makers' Association Handbook is now available. In addition to summarizing the aims, objects and activities of the Association, it also includes a complete list of the present 308 member-firms and buyers' guide index to their manufactures.

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CONTINENTAL MACHINE TOOLS

A Brief Survey of Recent Trends in Controls

IN many ways, Continental machine tool developments are following along similar lines to developments in this country. For example, machines are being made more rigid and more powerful to allow the fullest use to be made of the metal-removing properties of the latest cutting materials. The wider use of infinitely variable speeds and feeds has also been adopted to allow the optimum use to be made of the cutting tools.

Methods of controlling motions show considerable variation, but for straight-line motion, hydraulic infinitely-variable control gear appears to be the most popular, although the accuracy of mechanical gearing in conjunction with the simple adjustment of hydraulic gearing has been in great measure responsible for the development of hydro-mechanical mechanisms for hobbing machines and milling machines. There have also been considerable developments in the use of direct-current shunt-wound motors for obtaining infinitely variable speed regulation.

Considerable thought has also evidently been given to the reduction of non-productive time through quick tool changing, easy work handling, and single lever or push button control of the machine. Hydraulic, electronic and pneumatic devices are all used to effect economies in this field. While great use is still made of simple manually-operated mechanisms for indexing, there is increasing use of pre-selective and automatic sequence-controls which allow a whole complex work-cycle to be completed without any attention by the operator. Electric or electronic systems, combined with hydraulic gearing, have been successfully applied for lifting and clamping on radial drilling machines; the movement is initiated electrically and carried out hydraulically.

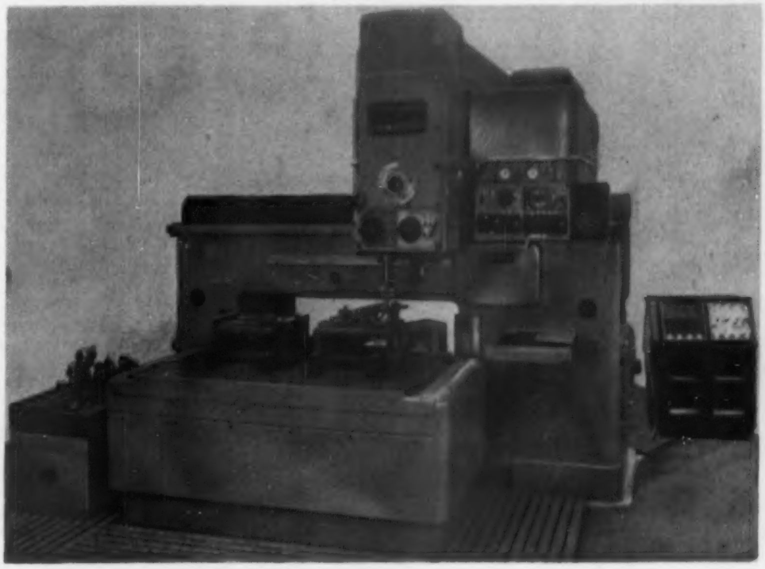

Fig. 1. BMAG boring machine
Rockwell Machine Tool Co.

In some ways, the most interesting machine tool development of the past few years is in automatic control from prepared data carried on a magnetic tape, paper tape or punched card. As is now well-known, this method of control completely eliminates the need for master templates or special cams. In addition, it is equally suitable for intermittent, short run or continuous production.

When the tape or punched-card technique is used, it is only necessary to insert the control medium into the machine, load the workpiece, adjust the tooling to a predetermined zero and then start machining. Generally, the operator does not even need to know the actual machining cycle. These control methods undoubtedly have great potentialities, but there are still problems to be solved in the preparation of the control data, particularly for complex forms.

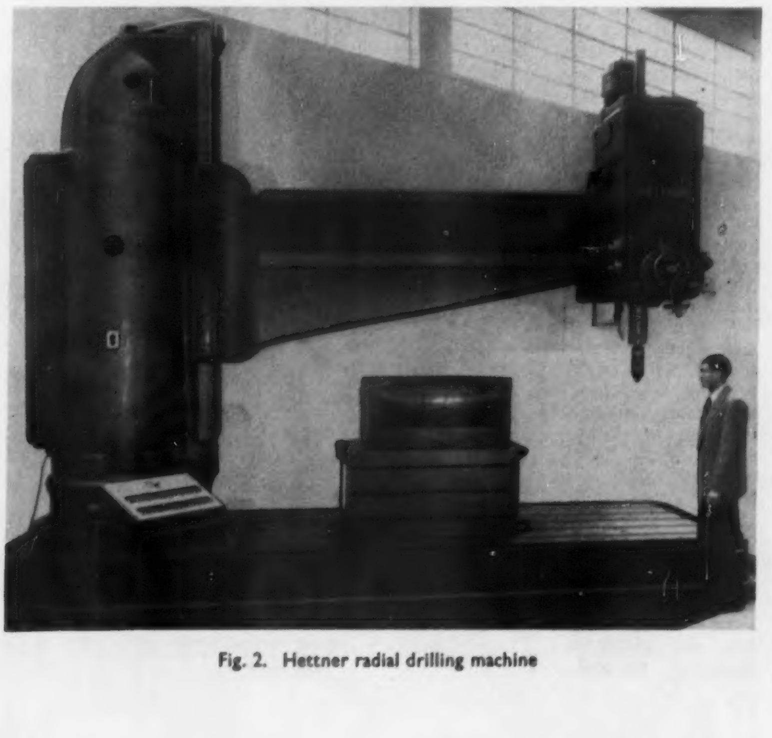

Fig. 2. Hettner radial drilling machine

Various methods of applying tape control have been adopted. For example, one machine incorporates a simple controlling system for machining relatively intricate details. It uses a 4½ in wide tape that will handle four co-ordinates with a maximum of 16 values for each co-ordinate; four tracks in the tape are used for each co-ordinate. In operation a small rocker mechanism is brought towards the tape and pivots according to the position of the holes. As it moves further it encounters a drive mechanism

Fig. 3. Lindner jig boring machine
Stedall Machine Tool Co.

that forces the rocker further in the direction in which it is moving. This movement closes electrical circuits that operate relays to provide any one of the 16 sets of voltages, and the selected voltage is fed to a synchronous motor to drive the machine slide.

A French machine has been developed on which the record is not made until the optimum machine cycle has been worked out. On this machine the operator can carry out a trial cut without making a recording. If this proves satisfactory, it can be recorded by push-button control; if unsatisfactory, further trial cuts can be carried out to determine optimum machining conditions before the recording is made.

A machine developed for the use of the punched-card technique is illustrated in Fig. 1. This is the BMAG fully - automatic co - ordinate boring machine by Berliner Maschinenbau-Aktien-Ges., Vormals L. Schwartzkoppf, which is handled in this country by the Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London, N.W.2. It incorporates a pre-selective system with repeater control. Through this method, a series of work-pieces can be machined in a controlled work-cycle, thus eliminating the loss of time and the possibility of introducing errors that are inseparable from manual re-setting of the machine for each hole.

The spindle head of the machine is mounted on a carriage for transverse and on a bridge assembly for longitudinal adjustment. When the machine is in use, the cross slides and longitudinal slides are locked hydraulically, and the table electrically. A hydraulic system gives an infinitely variable feed rate between 0 - 4 in/min and an automatic cycle for rapid approach feed and rapid return to the starting position.

For co-ordinate setting there are two groups of counter wheels, incorporating end measuring rods that allow settings to be obtained in increments of 0·0002 in in the two planes of location. Digits for the measurements are set in the dials which select the overall length of the unit measuring rods. The inch-measurement system allows six significant figures to be indicated in each group. In addition to provision for the pre-selection of co-ordinates, means are also provided to allow the speeds and feeds to be set for the next operation while a sequence is being carried out. When the spindle is retracted after an operation, a sequence control brings the next combination of end-measuring rods into position to control the table movement. This method of operation is usually adopted for one-off work. It is used in conjunction with a work card that lists the table movements in terms of co - ordinates and speeds and feeds for each station setting.

For producing a number of similar parts or for a work-piece that would require a large number of settings, the punched-card technique is employed. For this type of operation, the programme-selection cabinet shown at the right of the machine is used. The selection panel consists of a chest containing four sets of transverse and longitudinal setting points. Four location points can be obtained with each set and, when necessary, the four sets can be added.

The punched card is perforated in accordance with the prepared data. When a card is placed in position, jack plugs are inserted into the holes left uncovered by the holes in the card. The control lever is then moved to the automatic cycle position. The operation is initiated by depressing a push button when a signal lamp flashes to indicate that the machine has moved to the programme-point and is ready to be operated. When the first operation is completed, the second stage button is depressed, and so on until all the co-ordinates have been obtained. In its standard form this machine can be used for milling, drilling, boring and tapping work up to 50 in × 30 in and about 30 in height, with a maximum boring diameter of 10 in.

Another interesting form of automatic control has been applied to the radial drilling machine illustrated in Fig. 2. This machine is produced by Hettner, Bohrmaschinen - fabrik, F. Leber and Co., Munstereifel, Germany. It makes use of infinitely variable speed-and-feed gearboxes inter-related by pre-selection. The operator works at a control desk with two rows of 20 vertically rotatable wheels that carry numerical values arranged in column around their peripheries. Each pair of wheels in the upper and lower rows provides an operational setting for speed and feed. In this way, 20 pre-set values can be obtained at the start of

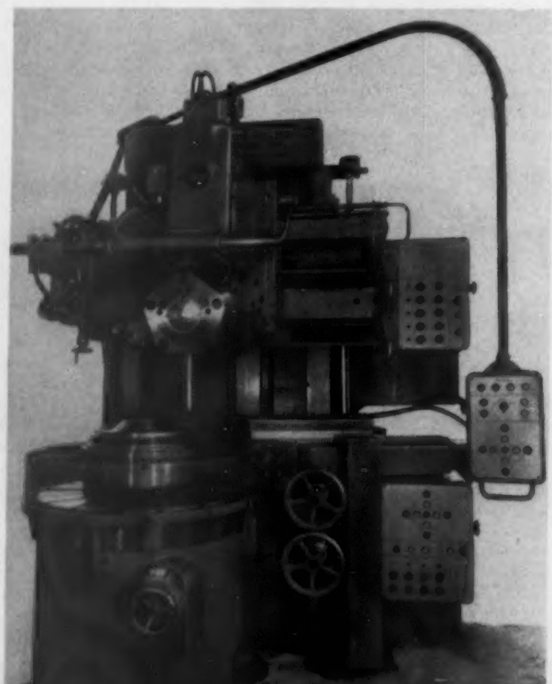

Fig. 4. Schiess vertical boring mill
Alfred Herbert Ltd.

an operation. The radial arm is operated by remote push-button control; fine positioning is obtained by means of servo motors. After a drilling cycle is completed, the next feed and speed may be indexed automatically if desired.

As the setting wheels are rotated, carbon brushes are moved across two fixed collectors inside the control desk to pre-set the feed per revolution and the spindle speed. Depending upon the segment of the collector at which the brushes stop, a certain combination of electro-magnets becomes operative to move the sliding change-wheels in the feed or speed gearbox. Actual preselection is then obtained by a rotary feeding commutator, which is actuated either by the stop-start lever or by the use of end-stops for each operating cycle. As a mechanical control is used for the speeds and feeds, the machine can be used for tapping.

Optical systems for setting the coordinates on jig boring machines are now well established, but refinements are still being introduced. For example, the Lindner machine illustrated in Fig. 3, handled in this country by the Stedall Machine Tool Co., 145/147, St. John Street, Clerkenwell, London, E.C.1, is fitted with a photo-electric cell for final adjustment of the work position. The basic method of positioning the work is by a micrometer with a double line straddling a millimetric graduation on a standard scale for verifying hundredths of a millimetre or microns of the coordinate. The photo-electric cell is used to simplify bringing the double line into position.

This machine also embodies the system of using finely engraved helices for measuring. By the use of a helix it is possible to obtain traverse in increments of 0·00005 in, using a vernier scale in conjunction with a graduated drum. The optical system allows the sighting point of the measuring scale to be projected and magnified on to projection screens in the control console where the image of the helical-scale-setting is centred between the straddle reference lines. To give extreme accuracy in setting, by eliminating parallax errors and necessity for interpolation by the operator, the projection system is augmented by the photo-electric cell centring device, which indicates the zero position when the image of a scribed line is exactly centred between the straddle reference lines. It is claimed that this device makes it possible to repeat settings to 0·00004 in.

For reducing non-productive time when selecting the transverse and longitudinal co-ordinates, the machine also incorporates a pre-selective system. In effect, it allows the operator, while one hole is being bored, to predetermine the next required position. The system comprises illuminated setting dials, which are used to obtain the integer millimetre value while the decimal values are obtained by means of a calibrated drum and vernier scale. After the start button is depressed, the table moves automatically to the vicinity of the pre-selected position; only a small correction through a micro-adjustment is necessary to bring the table into the final position. When the tables come to rest after traversing they are clamped automatically. The clamps are released by depressing the start button.

A particularly interesting development in the design of internal grinding machines may be found in a Jung Schleifmaschinen H. Graub, Berlin, machine, handled in this country by the Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London, N.W.2. This machine has been developed to take advantage of the latest practice in machine control. It is suitable for bores from $\frac{1}{4}$ to $1\frac{3}{16}$ in diameter and lengths up to $3\frac{5}{32}$ in.

Drive to the headstock is taken direct from a finely balanced motor, with the commutator forming part of the main spindle. Here, a pole-changing motor is used to allow three working speeds from 420 to 920 r.p.m. The headstock can be swivelled through 30 deg in the horizontal plane on either side of zero, but for automatic grinding the amount of swivel is limited to 10 deg.

During both rough and finish operations, automatic sizing is obtained by means of plug gauges; the sizing mechanism is used concurrently with the grinding motion. As soon as the plug gauge enters the workpiece, a limit switch causes the grinding cycle to stop. When the final size plug gauge enters the bore, the machine instantly returns the spindle to the starting position. The hydraulic and mechanical trips are operated from the front of the machine; cross traverse is controlled by hydraulically-driven cams.

Two simultaneously-operating feed systems, mechanically and hydraulically operated, are used. They independently affect the spindle feed. Compensation for wheel dressing is obtained by mechanical feed without altering the hydraulic setting. Manual in-feed and dressing in-feed are obtained mechanically in a manner which, while altering the position of the cross slide, does not affect the in-feed rhythm of the grinding cycle. The return traverse alters the position of the hydraulic in-feed to move the grinding wheel from the finish size position to the dressing position.

The operating cycle starts with a rapid traverse into the grinding zone with the cross slide in-feeding to the position for the roughing stage. At the completion of the roughing stage, the diamond holder moves into the dressing position, and final sizing correction gives automatic compensation. At the start of the finishing cycle, the spindle has rapid traverse to the in-feed position until grinding starts.

An interesting vertical turning and boring mill is shown in Fig. 4. It has been developed by Schiess Aktiengesellschaft, Berlin, for whom Alfred Herbert Ltd., Coventry, are the British agents. This machine has a working capacity of 39 in in diameter. It can be operated in the conventional manner, or through an electro-hydraulically controlled semi-automatic cycle. A feature of the design is that the automatic devices do not limit the scope of the machine for conventional applications. It is fitted with a side tool-head and a five-station turret on the cross rail. Both can be operated independently. The side clamps are applied and released automatically.

Feeds are infinitely variable from 0·002 in to 0·02 in per revolution, and there is infinitely variable fast traverse up to 120 in per minute. Setting and pre-selection are effected by switching and push button control in conjunction with adjustable dogs for the extent of feed-traverse. Taper turning can be carried out by using combined feed motions through change-wheel. By the use of suitable attachments, copy-machining, threading and scroll turning can also be carried out.

S.G. IRON

BY now it is well known that the undesirable characteristic of brittleness in cast iron can be overcome by transforming the graphite content of the material from flake to spheroidal form. This change is effected by treatment of the molten iron with magnesium. The main features of the process, and the properties of the spheroidal graphite cast iron made by it, are the subject of a 16 mm colour film, with sound commentary, entitled "S.G. Iron: Cast Iron That Bends." This film, which runs for 17 minutes, has been prepared recently by the Mond Nickel Co. Ltd., and is available for loan to technical societies, colleges and schools.

In the film, foundry production, including the magnesium treatment of the material, is illustrated. The properties that are typical of spheroidal graphite iron are demonstrated by shots showing bend, twist, impact, shock, tensile and machining tests, and by reference to some of the many applications in which this iron is being used. These applications include earth rammers, plough share points, gears, valves, pistons, bumper plates and other parts in which high strength and wear- and shock-resistance are essential. Pictures showing typical micro structures of grey and spheroidal graphite cast iron explain how the graphite flakes in the grey iron introduce stress raisers that give it its brittle characteristics, which are so undesirable in many applications.

FRICTION AND WEAR*

A Résumé of Current Knowledge

DETAIL measurements have shown that when two flat solid surfaces are placed together, the actual area of contact is very small: it varies with the load, and for flat steel surfaces it may be less than 1/10,000 of the apparent area[1]. This is because contact takes place only between the summits of the surface irregularities. The real area of contact is almost independent of the size of the surfaces. Moreover, it is but little influenced by their shape and degree of roughness.

Experiments suggest that the summits of the irregularities on which the bodies are supported flow plastically and are crushed until their cross section is sufficient to enable them to support the applied load. The real area of contact, A, is given by $A=W/P_m$, where P_m is the yield pressure of the metal and W is the load. This intense pressure causes adhesion, or pressure welding, and there is strong evidence that this is the cause of friction between sliding metal surfaces. The friction represents the force necessary to shear the junctions, and may be represented approximately by $F=As$, where s is the shear strength of the junctions.

When large flat surfaces are in contact, the points of contact are more widely distributed than when the areas of the surfaces are small. This is a simple explanation as to why friction is independent of the size of the apparent areas of contact. Moreover, the fact that the real area of contact is proportional to the load explains why the friction also is proportional to the load and the coefficient of friction $\mu=F/W$ is a constant. It is also to be expected that the friction of most metals, and indeed of most plastic solids, will have a similar value: a soft material gives a large A-value but a small s-value and *vice versa.*

In some instances the shearing occurs at the interface, but more frequently, because of work hardening of the contact region, it occurs a little distance away from it. Thus, a small metal fragment is generally detached from one of the surfaces and adheres to the other. This explains the wear phenomenon. The transfer of metal can take place from a hard metal to a soft one as well as conversely.

The coefficient of friction of metals is usually between 0·3 and 1·0. However, if the metals are placed in a vacuum and the surface films of oxide or adsorbed gas are removed, it is impossible to slide the metals together: the junctions spread until they are large in size, the friction rises to a high value and gross seizure occurs. If air is admitted, the friction falls again to its normal value.

Transfer of metal from cutting tools

From this background information, it is possible to develop ways of preventing the transfer of metal from cutting tools to the work piece. One of the simplest is lubrication, since oil reduces the amount of metallic contact and hinders the growth of junctions. Another is by avoiding, so far as possible, any slip between the surfaces. In normal circumstances, fragments are plucked out of both metals and transferred to the opposite surface. However, by controlling the relative hardness of the contacting surfaces, the transfer can be made to go predominantly in one direction.

Although the amounts of transfer are minute, in certain circumstances they may be detrimental as an impurity, for example, in the production of spectroscopically pure metals. Moreover, in some instances, these particles of a foreign metal may initiate a continuing damage out of all proportion to the apparent insignificance of the tiny fragment itself. Most metals owe their resistance to chemical attack to the layer of oxide on their surfaces. There is evidence that the transferred particles penetrate this oxide layer and adhere to the underlying metal. The presence of a particle of foreign metal, however small, may start a local electro-chemical corrosion and cause a break in the protective oxide layer. This can lead to large-scale continuing corrosion, particularly if the metal is in an adverse environment. If the metal component is subjected to permanent or transient stresses, local corrosion cracks may cause harmful stress concentrations.

Friction of non-metallic solids

There is evidence that the frictional behaviour of non-metallic solids is similar to that of metals. Molybdenum disulphide is a solid that has interesting frictional properties: it has a laminar plate-like structure similar to graphite and can have a low coefficient of friction. It is normally used in the form of a fine black powder, which is welded on to a steel surface. Since this only attaches it loosely to the surface, it may easily wear away.

An experiment has been carried out in which, instead of using a steel surface, a molybdenum surface was attacked chemically so as to form molybdenum disulphide *in situ*[2]. In these circumstances, it is firmly attached and, if the molybdenum is porous, it may penetrate to a considerable depth beneath the surface. Such surfaces can be heated without detrimental effects, and experiments show that even when the components are running red hot the coefficient of friction is still only 0·07, which is less than that of a well-oiled steel surface.

Some other studies that have proved interesting are on the frictional behaviour of some of the new plastics materials[3]. One of these is polytetrafluoroethylene, PTFE, which has an exceptionally low coefficient of friction. This low friction appears to be characteristic of the material itself, since even in a high vacuum, when extraneous surface films are removed, it is still found to be low. These frictional qualities are most desirable.

A study has also been made of the frictional behaviour of this material when incorporated in a porous metal, such as sintered copper[4]. The material produced in this way has the mechanical and thermal properties of sintered copper, but the frictional properties of the plastics, which is present as a thin film on the surface. The frictional characteristics of this combination are given in the table.

Coefficient of friction of metal impregnated with PTFE

Temperature deg C	Coefficient of friction, μ	
	Copper	Impregnated copper
15	About 1·0	0·05
100	About 1·0	0·05
200	About 1·0	0·05
250	Greater than 1·0	0·05

Surface temperatures

One method of measuring surface temperatures is to use the surfaces themselves as a thermometer. This may be done[5] by sliding two different metals together and measuring the thermo-electric potential developed. It is apparent that the electrical contact and the friction both occur at the same points, that is, where the surfaces touch, so the measurement gives information about the temperature of the surface layer of the metals at those points.

Such measurements confirm in a striking way the existence of local high temperatures at the points of contact of the rubbing surfaces. As would be expected, the temperatures reached depend upon the load, the speed of sliding and the thermal conductivity of the metals. An instrument of high frequency, such as a cathode ray oscillograph, must be used to record the tem-

** Abstracted from the 41st Thomas Hawksley Lecture, entitled "Recent Studies of Metallic Friction," by F. P. Bowden, Ph.D., Sc.D., D.Sc., F.R.S., delivered recently at a General Meeting of the Institution of Mechanical Engineers.*

peratures, since they fluctuate rapidly during sliding.

It is found that the melting points of most metals are readily reached and a further temperature rise does not occur. If the actual contact surfaces are the oxide films, whose melting point is high, much higher temperatures, of course, may be attained. If the melting points of the metals are high, temperatures of 500-1,000 deg C are easily reached momentarily. These temperatures are confined to the very thin surface layer, and the main mass of the metal appears to remain cool. The high temperature flashes, which in many instances are about 1,000 deg C, may last only for a few ten-thousandths of a second.

Other methods must be used to measure the temperatures between materials which are bad conductors of heat and electricity. One is the visual method[6]. If polished surfaces of glass or quartz are used and the apparatus is so arranged that a clear image of the rubbing surfaces can be seen, it is found that, in the dark, a number of tiny stars of light appear at the interface between them. The points of light are reddish in colour at low speeds, and become whiter and brighter as the speed or load is increased. It is clear that they correspond to small hot spots on the surface, and that their position and distribution over the surface change from instant to instant as the high points in intimate contact move or wear away, and as new points come into contact.

These hot spots can be photographed in a simple way. A photographic plate is placed on a turn-table with the glass side upward and allowed to rotate with a metal slider resting on it; the luminous hot spots developed at the points of rubbing contact photograph themselves on the emulsion. A third method can be employed. If one of the surfaces is of glass, quartz, or some other transparent material, the radiation from the hot spot may be transmitted through it and measured by a lead sulphide cell.

If the materials that are being rubbed together oxidize rapidly and if this oxidation is an exothermal process, as it is, for example, with aluminium and magnesium, the hot spot temperature may be much higher than the melting point of the material. This is because of the heat of oxidation liberated at the surface. Temperatures of 2,000 deg C or more may be reached.

These local hot spots are important factors in a number of physical processes associated with the rubbing of solids. They are, for example, very important in polishing. Strong evidence has been found that the formation of the polished, or Beilby, layer on metals and other solids when they are rubbed, is due to a localized melting or high temperature softening of the solid at the points of rubbing contact. The melted or softened solid is smeared over the surface, bridging and filling up the surface scratches, and is quickly cooled to form the micro-crystalline or "amorphous" Beilby layer. It has also been found that local hot spots play a major part in the initiation of explosions in solids and liquids when they are subjected to friction or to impact.

Friction

Experimental work has indicated that the low friction of snow and ice is due to surface melting produced by the frictional heating by sliding. At low temperatures, the static friction of ice is high. This is demonstrated by the fact that the friction of skis on snow does not fall to a low value until an appreciable speed of sliding is reached. The higher the temperature of the snow, the lower is the friction, and it is a minimum at 0 deg C.

The low friction obtained with PTFE, when applied to the underface of skis, is thought to be partly due to the inherently good frictional properties of this material and partly to the fact that it is not wetted by water. There is evidence that the frictional behaviour of different solids is influenced by the contact angle that the water film makes with the surface. With PTFE this contact angle is large.

The friction of outgassed graphite falls fairly steadily as the temperature is increased. It has been shown that at 2,000 deg C it is about one half of its value at room temperature. This is in marked contrast to the behaviour of a kindred lamellar crystal, molybdenum disulphide. The sharp rise of the friction of this material at 800-900 deg C is attributable to the decomposition of the sulphide resulting in a very high friction, normally associated with clean metals. The reason for the decrease in the friction of graphite at very high temperatures is not yet certain and further work is necessary to discover it, but it may be associated with the weakening of the Van de Wall forces at high temperatures and easier orientation during sliding.

Friction at high speed

A study of the behaviour of metals and other solids sliding at speeds of 1,000-2,000 m.p.h. has recently been made. The experimental methods employed to attain these speeds are of interest. A metal sphere is suspended by an electro-magnet and is illuminated, and the shadow falls on a photo-electric cell. If the ball sinks, the current in the magnet increases, if it rises, the current decreases, so it is held floating. It can be caused to rotate by a rotating magnetic field. If the sphere is in an evacuated glass chamber the forces opposing its rotation will be very small indeed and it will soon reach a very high speed, for example, a million or more r.p.m. The speed is limited only by the rupture strength of the sphere. As the top of the ball is illuminated, an etch mark on it can be detected by means of a photomultiplier cell and transmitted to a cathode ray oscillograph. The frequency of rotation can thus be measured accurately.

In the experiments, the fast-spinning ball rubs against three symmetrically arranged vertical surfaces. All three are initially in such a position that they cannot interfere with the ball during its acceleration. Two of them are mounted rigidly, and, when the desired speed is reached, a spring is released which pushes the third surface against the ball and thus moves it a very short distance sideways until it touches the two stationary surfaces. If the spring-operated surface is withdrawn, the ball returns quickly to its free-spinning position and can be slowed down in such a way that the high-speed wear marks are not further damaged. By measuring the loss of speed during a given sliding time under the known spring load, an average coefficient of friction is obtained.

In this way it is possible to measure the friction and to study the physical processes occurring over an enormous range of speeds, up to about 2,000 m.p.h. This work has only just begun. At normal speeds, the coefficient of friction of a steel sphere on a copper surface is about $\mu = 5$. This is to be expected, because the surfaces are clean and in a moderate vacuum. As the speed increases, the friction decreases: at 500 m.p.h., it is $\mu = 0{\cdot}9$. This decrease becomes more marked at higher speeds: at 1,000 m.p.h. it has fallen to $\mu = 0{\cdot}3$, and at 1,500 m.p.h. to $\mu = 0{\cdot}2$.

Abrasion due to sliding at a few feet per second is a common phenomenon. The damage at 500 m.p.h. is different in character and shows obvious signs of surface flow. At 1,200 m.p.h. the surface layers of copper approach the molten state, and thus a thin layer of lubricant is formed much in the same way as when a ski melts ice to form a layer of water on top. The viscosity of molten copper is of the same magnitude as that of water.

Although these results remain to be confirmed by further work, they lead to interesting conclusions. First, the frictional resistance to the sliding of metals at these very high speeds can be much less than at normal speeds. Second, the wear per unit of distance slid can be much less. Perhaps the technical problems involved in sliding metals at these enormous speeds are not so difficult as at first might be supposed.

Rolling friction

The source of free rolling friction, which is generally less than 0·001, has long been a matter of speculation. When a hard steel ball rolls in a groove, points on a diametral plane through the centre of the axis of rolling, that is, on the equator, measure out on the track a larger distance per revolution than points nearer to the poles, therefore, some slip presumably must occur. Heathcote[7], who first pointed out this fact, attributed a major part of the rolling resistance to this differential slip. However, experiments with balls rolling in rubber grooves show that differential slip of this type occurs only when the groove is exceptionally deep. Apparently, the main losses in shallower grooves are due to elastic hysteresis in the rubber. It would appear that the hysteresis loss mechanism accounts for

the experimental observation that the rolling friction is essentially the same whether the surfaces are clean or lubricated.

If a copper ball is rolled over an identical copper ball the region of contact is, by symmetry, plane, so the Heathcote slip is eliminated. Again, stretching of the interfacial elements will be the same on both balls, so no slip due to differential stretching of these elements will occur. Tabor's experiments[8] show that the rolling resistance of the two balls is independent of the presence of lubricants. It is about the same as the rolling resistance of a hard steel ball in a copper groove, and involves elastic hysteresis losses that may be as great as 20 per cent. From these results, it is clear that rolling friction does not generally arise from interfacial slip. Thus lubricant films may play an important part in reducing surface attrition or wear, but have little effect on the rolling resistance itself.

In practice, the behaviour of ball and roller bearings is complicated by friction of the cage and other factors, but it is clear that the hysteresis losses must play a very important part. These losses have a marked influence on the fatigue life of bearings. The work of Tabor and others may have practical applications in the search for bearings of lower rolling resistance and longer life.

Conclusion

In engineering practice, of course, the friction is a major problem. It is not so much the work that has to be done in overcoming friction that is important, although this can be considerable. In a modern motor car, for example, about 20 per cent of the power is wasted in overcoming friction, in an aeroplane piston engine it is about 9 per cent and in a modern turbo-jet about 1½-2 per cent. The real trouble is the damage that is done by friction; that is, the wear or seizure of some vital part of the machine. It is this factor, perhaps more than any other, that limits the design and shortens the effective working life of a machine.

Much ingenuity has been exercised on this problem and, up to a point, it has been successful, but only up to a point. One of the greatest limitations is temperature. It is difficult at present to run bearings hotter than 300 deg C. This is now becoming a serious limitation. To-day, the engineer would like to run bearings and to slide surfaces at 1,000 deg C and at very high speeds, but he cannot do this until new and unconventional lubricants and surfaces have been developed. A fuller and more detailed understanding of the physics of the sliding process under these extreme conditions will help. The problem, particularly with the advent of atomic power, is an urgent one.

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FLAME HARDENING OF CRANKSHAFTS

FLAME hardening of 6- and 8-cylinder truck engine crankshafts has been adopted by the Chrysler Corporation, it is reported by H. Chase in *Automotive Industries,* October 15, 1954. The S.A.E. 1046 steel crankshafts are first furnace hardened to 228-269 Brinell and subsequently rough machined. They are then flame hardened in Cincinnati Flamatic units which produce a hardness of 55-60 Rockwell C at the centres of the bearing surfaces, the case extending to points ⅛ in from the crankshaft fillets. Heating is effected by an oxygen-propane mixture, the flow of which is controlled so that temperatures are highest at the centres of the bearing areas. Times of heating are automatically controlled and oil holes in the bearing surfaces are plugged to avoid overheating their edges.

In the flame hardening process, six-throw shafts are loaded between centres in a horizontal air clamping and indexing fixture which positions the shaft during heating. On reaching the heating station, the fixture is advanced so that the burner blocks approach the bearing surfaces. Heating takes place for 33 sec while the shaft is rotated. Burners heating the crankpin surfaces reciprocate to retain their position relative to these surfaces as the shaft rotates. At the end of the heating period, the shaft is lowered by rocker arms into a water quench tank. Total time per cycle of the machine is about 80 sec. Shafts are then examined for hardness.

In the hardening machine for the shafts of 8-cylinder engines, which employs a natural gas/oxygen fuel, main bearing surfaces only are first heated and spray quenched, and each crankpin surface is then individually heated and quenched. Since the burners do not reciprocate, the shaft must be set in five different positions to enable all bearing surfaces to be heated.

After hardening and inspection, the crankshafts are tempered for 4 hr at 450 deg F. Case depth is a minimum of 0·125 in. It is stated that the process has proved successful, and that the results made possible by flame hardening have not been attained by other means. *M.I.R.A. Abstract No.* 7052.

A NEW QUARTERLY

THE first issue of a new quarterly, *Materials Handling News,* which is being produced by Associated Iliffe Press, will appear on 1st July, 1955. This publication is intended for circulation among firms making less than the optimum use of mechanical aids for handling materials; this is a market quite distinct from that already covered by the parent journal *Mechanical Handling.*

Brief and factual editorials will be planned to arrest the attention of readers. The inclusion of a pre-paid card keyed to editorial items and advertisements, is expected to stimulate continued interest.

Each quarter 20,000 copies will be distributed free and under careful control to senior executives in appropriate organizations. It is expected that about 10,000 will circulate in the firms to which they are addressed, to give a total readership in the order of 50,000, and of these 25 per cent will be overseas.

Materials Handling News will have a modern newspaper format, with an attractive two-colour cover and a 17¼ in × 11⅞ in page size. Publication dates will be 1st July, October, January and April.

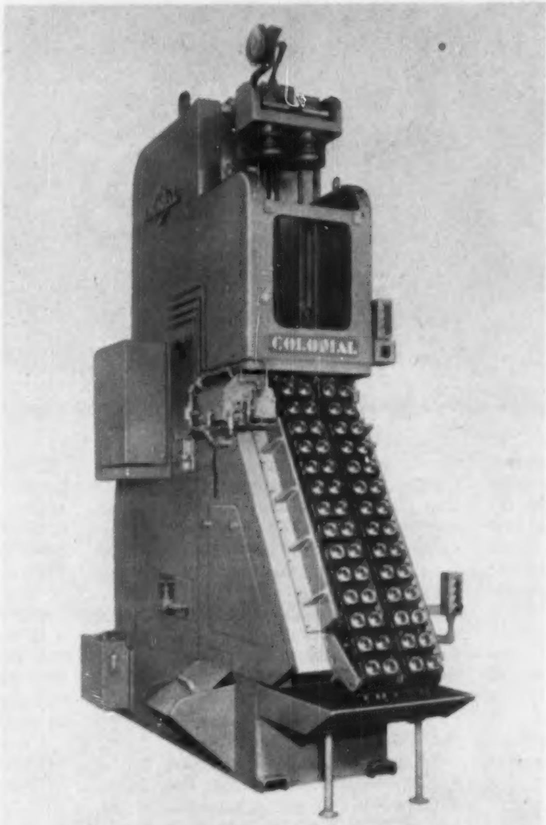
As this new publication will have behind it the experience and knowledge of the editorial staff of *Mechanical Handling,* it can confidently be expected to be of great interest to firms, both at home and abroad, aiming to increase productivity.

AUTOMATIC BROACHING

An American Development for High Production

DURING the past year the automobile industry in the United States of America, in conjunction with the makers of the necessary equipment, has made remarkable advances in production techniques. In the main, the developments have all had one common aim: increased production from a smaller labour force. To attain this aim, it has been necessary to adopt completely new conceptions of the methods for handling work between operations and for loading and unloading machines. Briefly, it has been necessary to mechanize these functions so that they are automatically performed by mechanisms instead of manually by operators.

One of the most interesting and unusual of recent developments is one applied by the Colonial Broach Company, Michigan, U.S.A., to a machine for broaching the splines in two different models of a differential side gear at a total rate for both gears of 800 per hour. The gears are fed to four, side by side, broaching stations by means of a continuous chain loading conveyor carrying 144 individual work-holding fixtures mounted four abreast. Conveyor travel begins at a convenient height for manual loading and continues up at a steep angle until the broaching level is reached. At this point the path of the fixtures levels off horizontally to accommodate the broaching and broach return strokes. A standard Colonial RU 15-48 pull-up broaching machine was adapted to this interesting operation. It is shown in the accompanying illustration.



Colonial broaching machine with conveyor loading

A complete cycle comprises:—

(1) Four gears are indexed into the broaching position.
(2) The splines are broached.
(3) The fixtures are indexed for broach return.
(4) The broaches are returned through gaps between the work-holding fixtures.

This cycle takes only 18 seconds. Small cams on the backs of the fixtures actuate the limit switches for controlling the cycle sequence.

As the conveyor indexes past the broaching station, the broached parts are carried over the top of the conveyor travel. Then, as the conveyor starts on its downward run, the gears drop out of the fixtures through two separate chutes in each side of the machine. This simple gravity-type system of unloading makes it an easy matter for the operator to make a continuous check of each individual broaching operation.

The conveyor is indexed hydraulically. To ensure positive indexing, a second hydraulic cylinder locks each group of fixtures in place for broaching. Forward and reversing clutches eliminate any backlash or over-ride during the indexing motion. It is interesting to note that clamping fixtures are not required. Instead, stationary hold-down bars above the broaching station keep each gear seated securely in its fixture as the broach is drawn through the hole.

The two gears for which this machine has been developed differ in hub depth, diameter and spline size. One requires a 16-tooth involute spline with a P.D. of 1·1562 in, and the other a 10-tooth involute spline with a P.D. of 1·125 in. To keep the two types separate, the two right-hand conveyor rows are tooled for one model, and the two left-hand rows for the other.

Gaston E. Marbaix Ltd., Devonshire House, Vicarage Crescent, London, S.W.11, are the sole British agents for these machines.

CORRESPONDENCE

HARD CHROMIUM PLATING

SIR,—With reference to the article on page 66 of the February issue of *Automobile Engineer*, may we point out that the impression given, that this is an entirely new technique not previously available in this country, is incorrect.

Monochrome Limited have specialized in the direct application of hard chromium to light alloys for many years, and during the war many thousands of components were so treated. Of recent years, the field has been widened to include engine cylinders and liners finished by our "Honeychrome" process with very satisfactory results, in fact, one particular engine maker has to date installed over ten thousand cylinders of this type.

In addition to our own production, the process has been licensed both in Europe and North America, and it can be safely said that the advantages as pointed out in your article are actually attained in practice.

It may be of interest to note that in addition to the normal materials used for engine liners, non-ferrous metals other than aluminium alloys have been tried for particular applications, and in these cases also, we are able to hard chromium plate directly.

B. TURNER.

Monochrome Limited.

GREGOIRE SUSPENSION

Some Aspects of the Application of this System to Public Service Vehicles

TO maintain a constant frequency in suspension systems, it is necessary to have springing which varies in rate in direct proportion to the load carried. Variable rate springing has another advantage in that it reduces the tendency to resonant vibration; therefore, smaller shock absorbers can be employed and road holding is improved. With a constant rate suspension designed to suit full load conditions, the natural frequency of the system increases as the load is reduced. In these circumstances, not only is the ride not so comfortable, but the tendency for fatigue failures to occur is also increased.

Gregoire assister spring installation on a commercial vehicle chassis

Most of the variable rate suspension systems that have been in use for many years take the form of a leaf spring arrangement, giving two, three or more stages, and a variable rate can also be obtained by using torsion bar springs with short lever arms. However, the well-known Gregoire arrangement is different in that separate coil springs are employed to give the variable rate. Moreover, it is continuously variable instead of variable in separate stages, as are the leaf spring systems. The sole licensees for the Gregoire suspension in this country are William E. Carey Ltd., of Redbank, Manchester 4.

Public service vehicle manufacturers and operators have recently become increasingly interested in this system. Operators of these vehicles are now finding it necessary to direct efforts towards attracting passengers who otherwise would travel by private car. To do so it is necessary, among other things, to provide suspension systems that give riding characteristics that are comparable, so far as comfort is concerned, with those of private cars. This is difficult with conventional springs because of the great difference between the fully and lightly laden conditions, and in some instances the problem is further complicated by restrictions imposed by the need for maintaining adequate roll stiffness. One solution to these problems is the adoption of the Gregoire arrangement.

Originally the Gregoire coil spring system was designed for use alone, that is, without any other springs. In its simplest form, it comprises a coil spring, one end of which is pivot mounted on the frame and the other on the axle. In the unladen position, the axis of the spring is more or less horizontal so, as the axle deflects vertically and the spring axis is inclined more and more from the horizontal, the spring rate is increased. Movement of the axle is controlled by a conventional link system.

Although this simple Gregoire arrangement is superior to a constant rate suspension, it is difficult, and in many cases impossible, to maintain a constant frequency under all conditions of load. Subsequent development work revealed that the best arrangement is to employ a conventional spring to carry the unladen weight of the vehicle, and to use the Gregoire springs only to maintain a constant rate as the load is increased.

When the Gregoire system is used in this way, the springs are termed *corrector springs*. In some instances, notably with heavier types of vehicles, one corrector spring is not enough, so a second is employed. Normally these springs are installed one in front and the other behind the axle, with their axes more or less in line, as in the accompanying illustration. In designing the system, it is necessary, particularly when only one corrector is employed, to ensure that the turning moment about the leaf spring seat on the axle is as small as possible.

In general, it is not always practicable to obtain the ideal solution, because of restricted space available. The correctors, for example, may have to be designed to carry the maximum load possible within the limits imposed by the installation, and the main springs have to carry the remainder. However, the range of frequencies that will give comfortable riding qualities is fairly wide, so satisfactory results can usually be obtained.

MOLLERISING

MOLLERISING is an economical method, developed in Sweden by G. A. Moller, of coating steel with aluminium to protect it against corrosion or high temperatures. It overcomes difficulties experienced in earlier processes caused by: (1) the highly solvent properties of molten aluminium; (2) the high cost of external heating of the container of the molten metal; (3) oxidation of the aluminium; (4) contamination of the aluminium by dissolved steel and by the flux employed in prior treatment of the work.

The new process, described by H. E. Linden in an S.A.E. Preprint, January 1954, utilizes an electric salt bath furnace such as commonly serves for the heat-treating of steel. A mixture including barium chloride and sodium chloride is melted in the furnace, and on the top is a float of molten aluminium some 2 in thick. A thin surface film of salt, formed from the salt vapours, prevents oxidation. Metal consumed during plating is made good by an ingot of aluminium suspended in the salt.

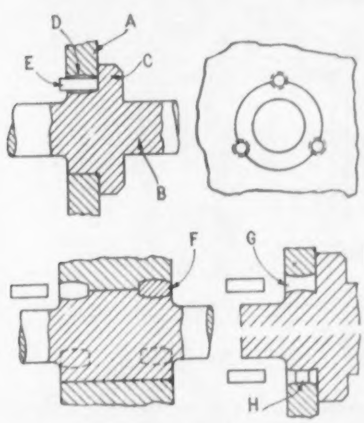
The work, thoroughly cleaned and pickled, and suspended on a rod or wire, is immersed in the salt through the molten metal. After having attained the same temperature as the salt, it is withdrawn through the molten aluminium. Addition of a flow of low-voltage direct current is said to improve the coating process. A layer of iron chloride is formed on the steel and, when the latter enters the molten aluminium, a reaction occurs between the chloride and the aluminium. The layer of very fine, active molecular iron so created is fully wetted by the aluminium, and the iron-aluminium alloy forms a strong bond between the metals. The interface produced with the current is almost twice as thick as when no current is applied, but is very ductile. Withdrawal of the steel at 150 ft/min through the 2 in of molten aluminium gives a contact lasting ¼ sec, a very brief period of contact between steel and aluminium ensuring enhanced ductility of the coating. Mollerising can compete in cheapness with galvanizing; it has been used on titanium and molybdenum. *M.I.R.A. Abstract No.* 7076.

CURRENT PATENTS

A Review of Recent Automobile Specifications

Securing wheels or rotors to shafts

IN certain machine constructions it may be necessary to secure a wheel on a shaft against radial and axial displacement. Where the wheel or rotor is of a hard or brittle material, difficult to machine and having only a small elongation, the conventional methods of shrinking, keying or splining cannot conveniently be employed. The difficulty is overcome by forming, at the junction diameter, barrelled or waisted sockets into which metal pegs are inserted and upset to contact firmly the recess walls in both elements. Should the wheel be of a compacted metal powder the recesses



No. 709619

may be formed therein before sintering.

Wheel A is registered on a shaft B and located by a flange C. Three barrel-shaped sockets D are formed half in the wheel and half in the shaft. Suitably heated pegs E are inserted in the sockets and by means of a press are upset to fill the sockets. Where the contact area between the two parts is greater and axially extended, pegs may be fitted at both end faces, as at F. This method is suitable for armouring rollers with sleeves of hard material.

Instead of being barrelled the sockets may be waisted, as shown at G. An alternative form H, comprising a cylindrical centre portion with two coned end portions, is simpler to machine. *Patent No.* 709519. *Metro-Cutanit Ltd.*

Abrasive blasting of light alloy components

IT is common practice to obliterate the scars left by flash metal, sprues and risers, and to impart a uniform surface finish to light alloy castings by abrasive blasting. Quartz sand, iron shot or steel shot projected by compressed air or centrifugal impellers is widely used as the abrading medium. Each of these materials, however, entails certain disadvantages. Sand blasting presents a serious dust hazard to the health of the operator, and also the grains of sand rapidly break down on impact and lose their effectiveness. Iron or steel shot or grit leaves on the casting a thin film of iron dust that can be removed only with great difficulty. If it is not removed it will inevitably result in the formation of rust when the component is exposed to atmospheric moisture.

The invention proposes the use of a light alloy shot or grit to avoid such disadvantages. By selection of the light alloy for the shot an appropriate hardness can be obtained, and it is possible to reduce large-grained shot or lathe swarf in crushing equipment to produce angular grit of smaller size.

Another alternative is to use drawn wire chopped into lengths once to twice its diameter. An alloy having a percentage composition of copper 16·5, manganese 0·4, iron 1·4, silicon 0·3 and the balance aluminium is well suited to the purpose. *Patent No.* 709869 *Georg Fischer A.G. (Switzerland).*

Mounting head linings

TO permit prefabrication of head lining panels and to reduce the time required to assemble in the vehicle body, it is proposed to secure the fabric lining material to a light metal framing which is sprung into position to seat on a rebate provided on the upper edges of the interior finisher panels. In the example unitary front and rear panels abut on a transverse line x—x.

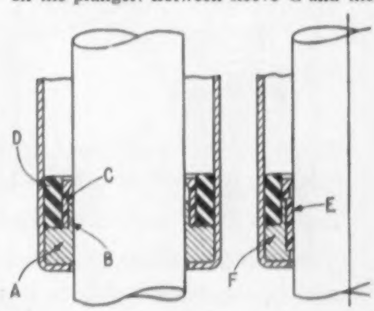
The upper edge portions of the finisher panels A are formed with right-angled flanges B and C defining a rebate all round the interior of the body. Each lining panel consists of a light skeleton frame of beaded-edge metal strips. The lining material D is secured to the peripheral strip and at suitable points of the longitudinal and transverse strips E and F.

These panels are mounted by bowing them from a substantially flat condition and allowing them to spring back against the side C of the rebate. They are retained in a slightly bowed condition and no extraneous means of fastening is required. *Patent No.* 710131. *Austin Motor Co. Ltd.*

Packing device

THIS sealing arrangement for the plunger of a telescopic shock absorber has a relatively soft resilient ring which is compressed on assembly to constrain initially a relatively hard, resilient ring engaging the plunger. In operation, fluid pressure in the cylinder further compresses the softer ring and thus increases the radial pressure on the harder ring. The device is claimed to offer a low frictional resistance to sliding movement and to be noiseless in operation.

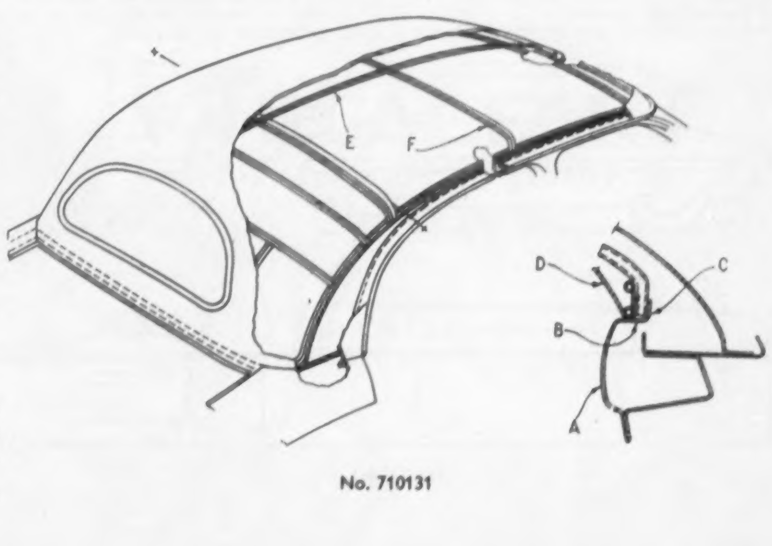
Referring to the first example, the cylinder has an inturned flange supporting a metal abutment ring A formed with a thin rigid extension B. On this is fitted a sleeve C of nylon having, at its inner end, an internal circumferential flange of triangular section, the apex of which bears on the plunger. Between sleeve C and the



No. 710374

cylinder bore is compressed a rubber ring D, seating on ring A and extending inwardly slightly beyond the end of sleeve C.

The alternative arrangement has the nylon sleeve E mounted inside abutment ring F and bearing directly on the plunger. At its inner end it is internally recessed and formed with a terminal triangular flange, as in the first example. *Patent No.* 710374. *Automotive Products Co. Ltd. and Levitation Ltd.*

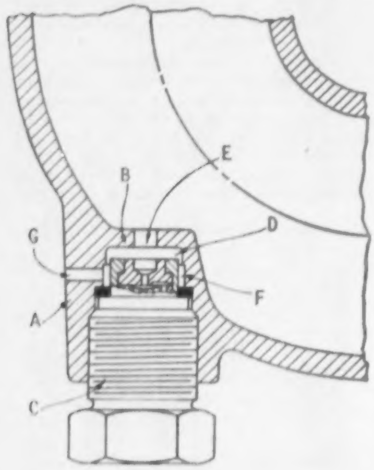


No. 710131

Petrol injection

FOR petrol engines of the type in which the fuel is injected into the individual branches of the intake manifold, it is an economy to use a rotary pump instead of a high-pressure piston pump of a construction derived from the orthodox fuel injection pump for diesel engines. With such relatively low-pressure pumps, however, the pressure available at the lower range of engine speeds, particularly when idling, is insufficient to secure satisfactory atomization of the fuel. Consequently, air and fuel mixing is inadequate, vaporization is poor and the rate of fuel consumption is increased beyond the optimum value. The invention aims to obviate this disadvantage.

The induction branch is bossed at A



No. 710213

and formed with a recess, closed internally by the wall B, to accommodate the injection nozzle C at a specific depth so that its end face is spaced from wall B to leave a disc-shaped chamber D. In wall B is an axially aligned port E of substantially greater diameter than the discharge orifice of the nozzle. Chamber D is in communication, by way of the clearance between the nozzle end and the wall of the recess, with an annular chamber F. By means of drillings G this chamber is open to atmosphere or, in the case of supercharged engines, is connected to the pressure pipe of the charging system.

As a result of the depression in the intake manifold, atmospheric air or pressurized air from the supercharger will flow into chamber D and thence, together with the fuel, through port E into the induction branch. This preliminary mixing of air and fuel is claimed to ensure a satisfactory combustible mixture even at low engine speeds. *Patent No.* 710213. *Ustav Pro Vyzkum Motorovich Vozidel and J. Bottger (Czechoslovakia).*

Hard-chrome plating

THIS invention is concerned with the chromium plating of metal objects, particularly of steel shafts, that are required to be plated only in specific areas. It is customary to "stop off" areas to remain unplated by means of a coating of lacquer but this procedure is not entirely satisfactory. Extreme care is necessary to obtain adequate protection and, due to faulty adhesion, porosity, or blistering of the lacquer, an unsightly straying of the chromium layer may occur at the transition boundary defining coated and uncoated areas.

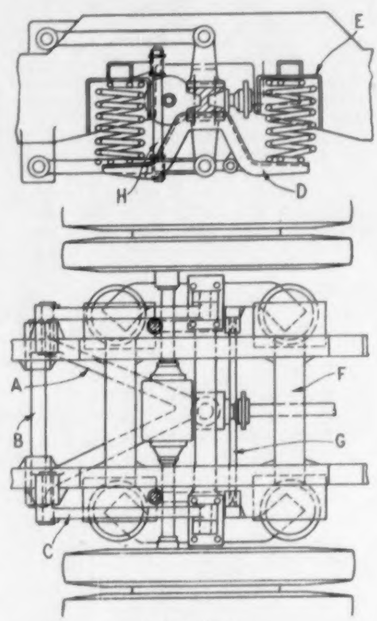
It is proposed to mask the area not to be coated by covering it with a conductive metal foil. Preferably a strip of aluminium foil is wound on the shaft. Thus, the layer of chromium is deposited over the foil as well as over the area to be coated and it is not possible for stray fields to be produced at the margins. When the masking strip is removed, there remains a uniform layer of chromium having a clearly defined boundary.

In a further development of the method a layer of lacquer terminating at a distance from the transition boundary is applied to the foil covering. The critical zone is thereby transferred from the boundary and on to the foil where the occurrence of stray field phenomena can do no harm. *Patent No.* 710548. *Gebr. Schoch Hartchrom G.m.b.H. (Germany).*

Streamlined body construction

THIS body structure comprises two hollow side portions A, of elliptical section and enshrouding the road wheels, flanking a middle portion B including bonnet, passenger space, and boot. By laterally inclining parts A, preferably at the camber angle of the wheels, encroachment on the width of the passenger space is minimized. The body part B may have a flat floor C disposed at the height of the bottom edge of the sides, or the floor may be bowed in relation to the ground in order to increase the clearance.

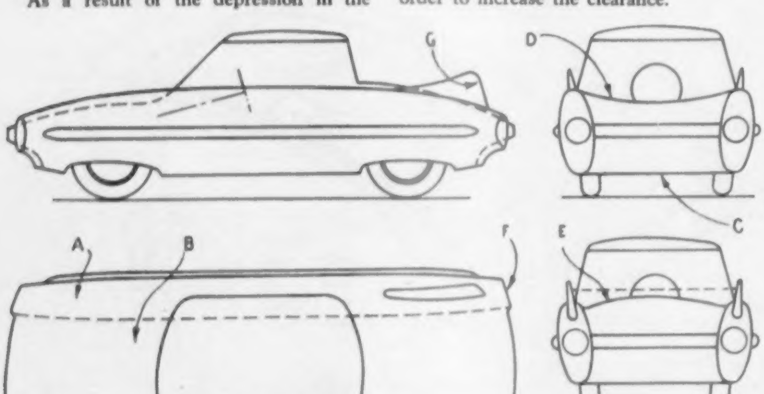
To give a good field of vision from the centrally positioned driving seat, the top wall D of part B forward of the passenger space presents a concave surface upwardly; the top wall E of the boot is convex. Each side portion may consist of a central part of substantially rectilinear shape with tapered end parts at front and rear. These tapered ends terminate in circular apertures F on, above, or below the longitudinal axis of the side member. They are



No. 710058

closed by discs or, presumably, occupied by head and tail lights.

At the rear ends of parts A upstanding stabilizing fins G are provided and between these a radio aerial may be extended, as indicated in broken line. As alternatives to the elliptical section of the side members, tear-drop or egg-shaped sections may be used, with the narrower end of the section directed either upwards or downwards. *Patent No.* 710463. *Daimler-Benz A.G. (Germany).*



No. 710463

Bus rear suspension

IN this arrangement, the axle is guided by longitudinally disposed parallel links and sprung by pairs of helical springs, one spring of each pair being forward and one rearward of the axle. The drive is through an axle-mounted differential gear and half-shafts to hub-type reduction gears.

A triangulated member A, pivoted on a transverse shaft B and connected by a ball joint to a bracket depending from the centre of the axle, constitutes the lower element of the parallel linkage. The upper lateral links C are either rubber-bushed or ball-jointed at their ends to permit the axle to tilt about the lower ball joint. Springs are mounted between the ends of stirrup-type members D, secured to the axle beam, and frame brackets E in way of frame cross members F. A torsion-rod stabilizer G mounted in lugs on members D has forward extending lever arms connected to vertical rods H bracketed to the frame side members. Connections at both ends of these rods are rubber-bushed. *Patent No.* 710058. *Daimler-Benz A.G. (Germany).*

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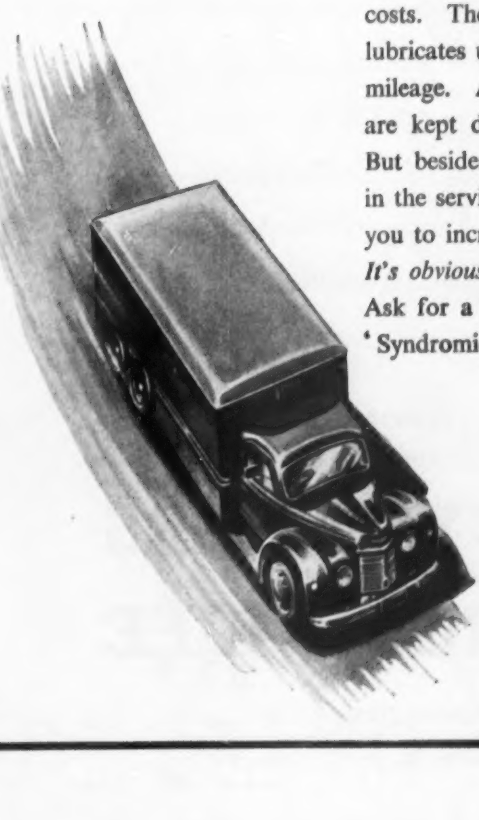

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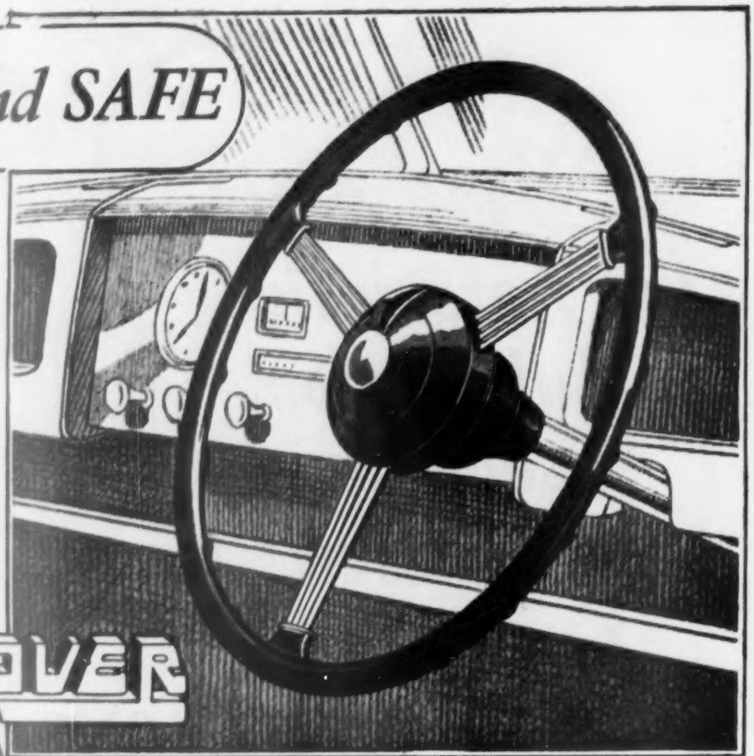

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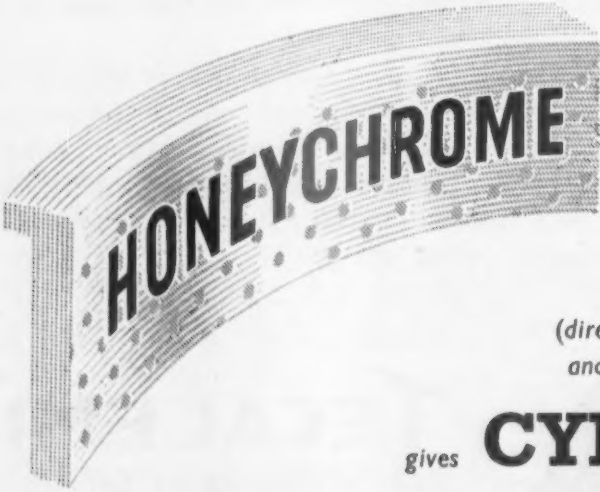

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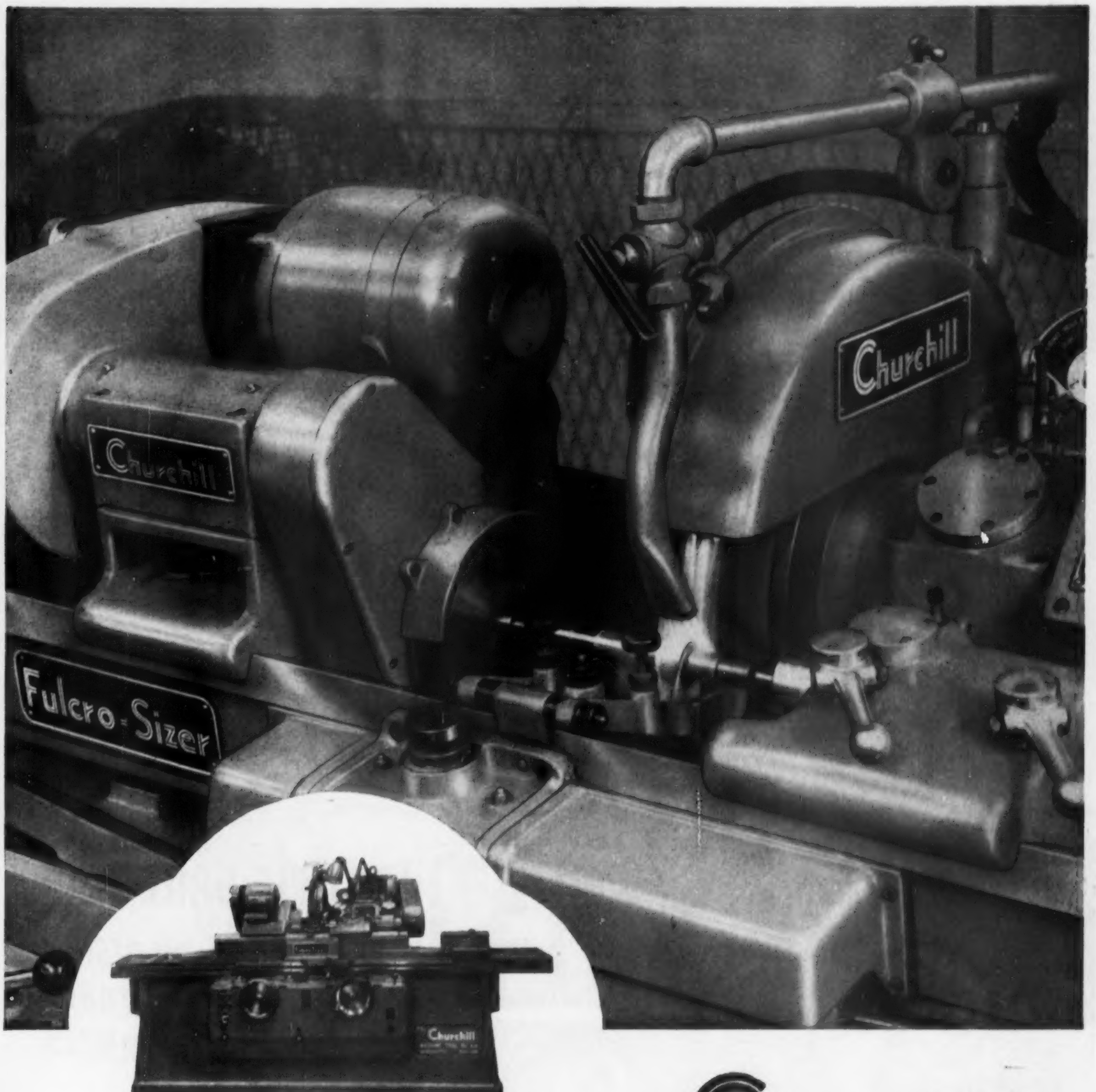

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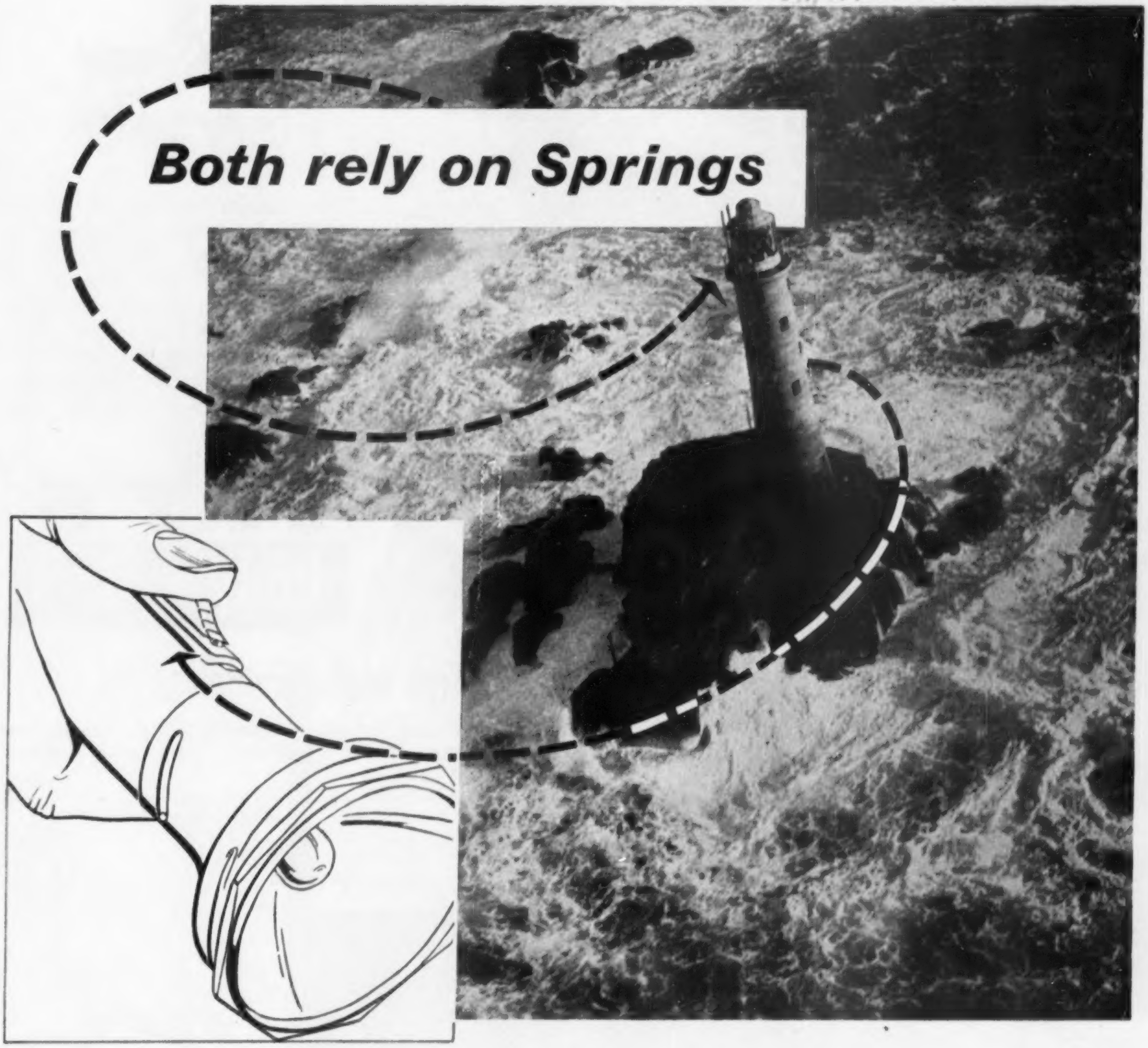

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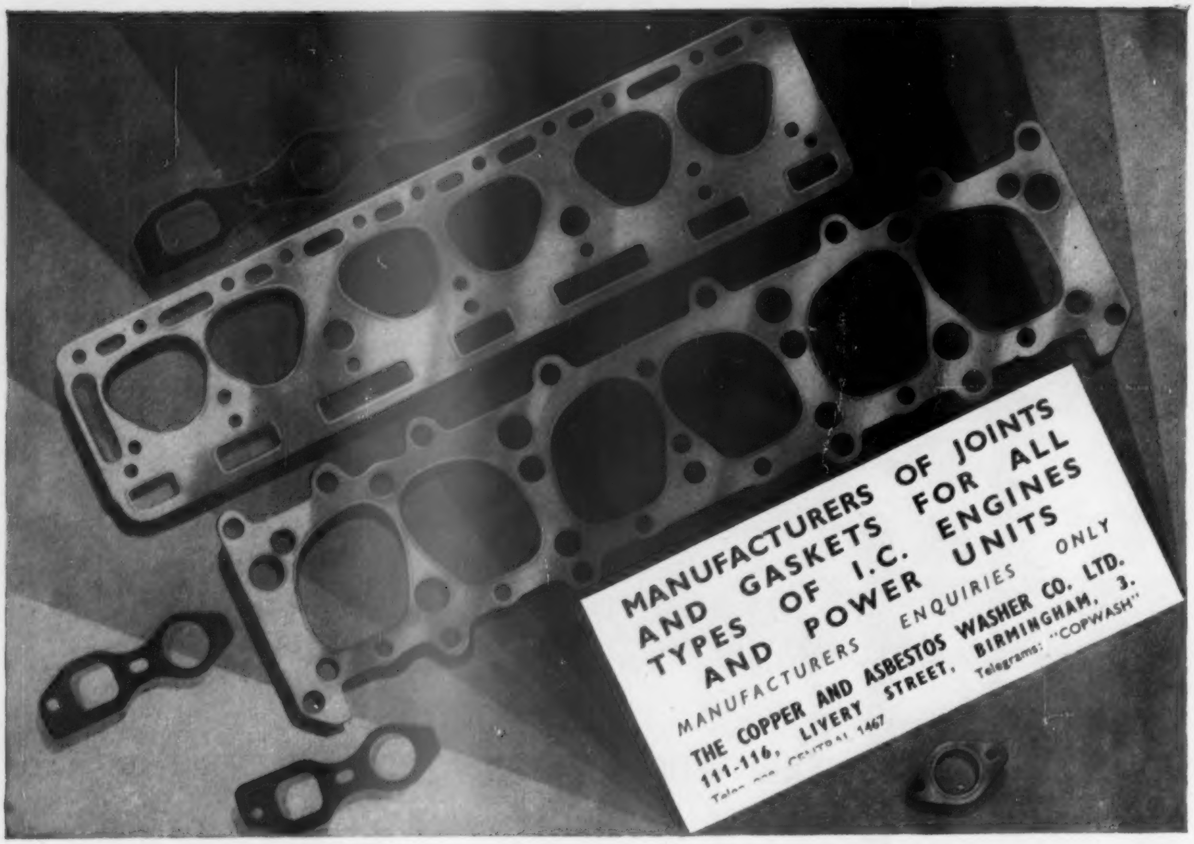

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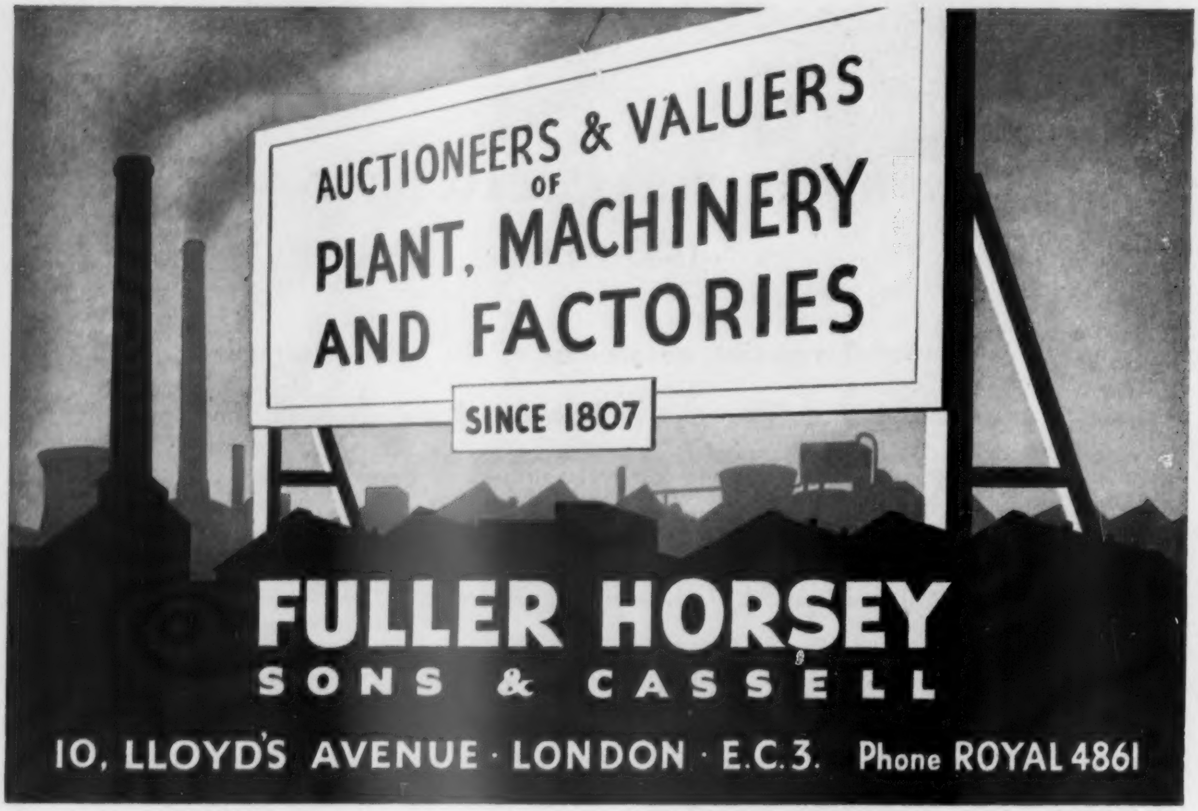

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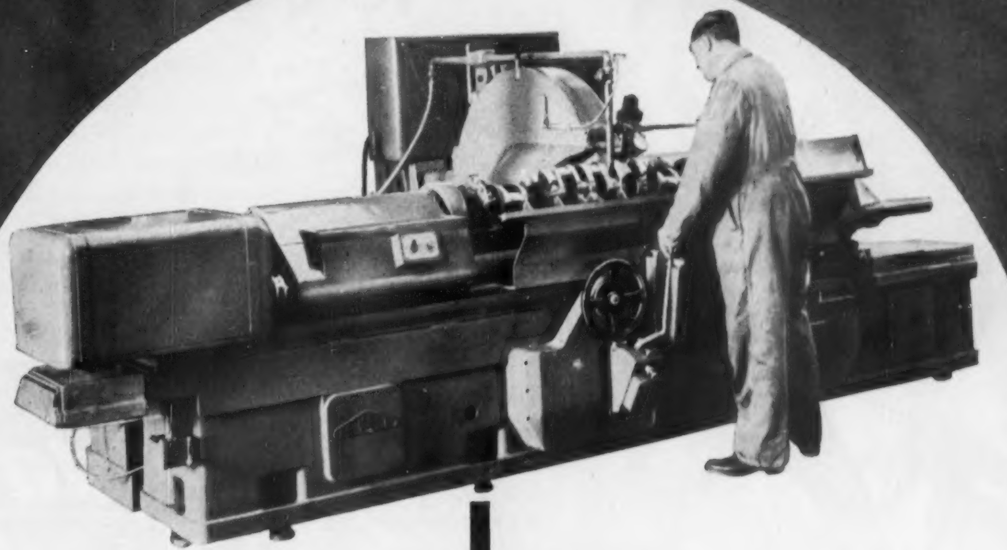

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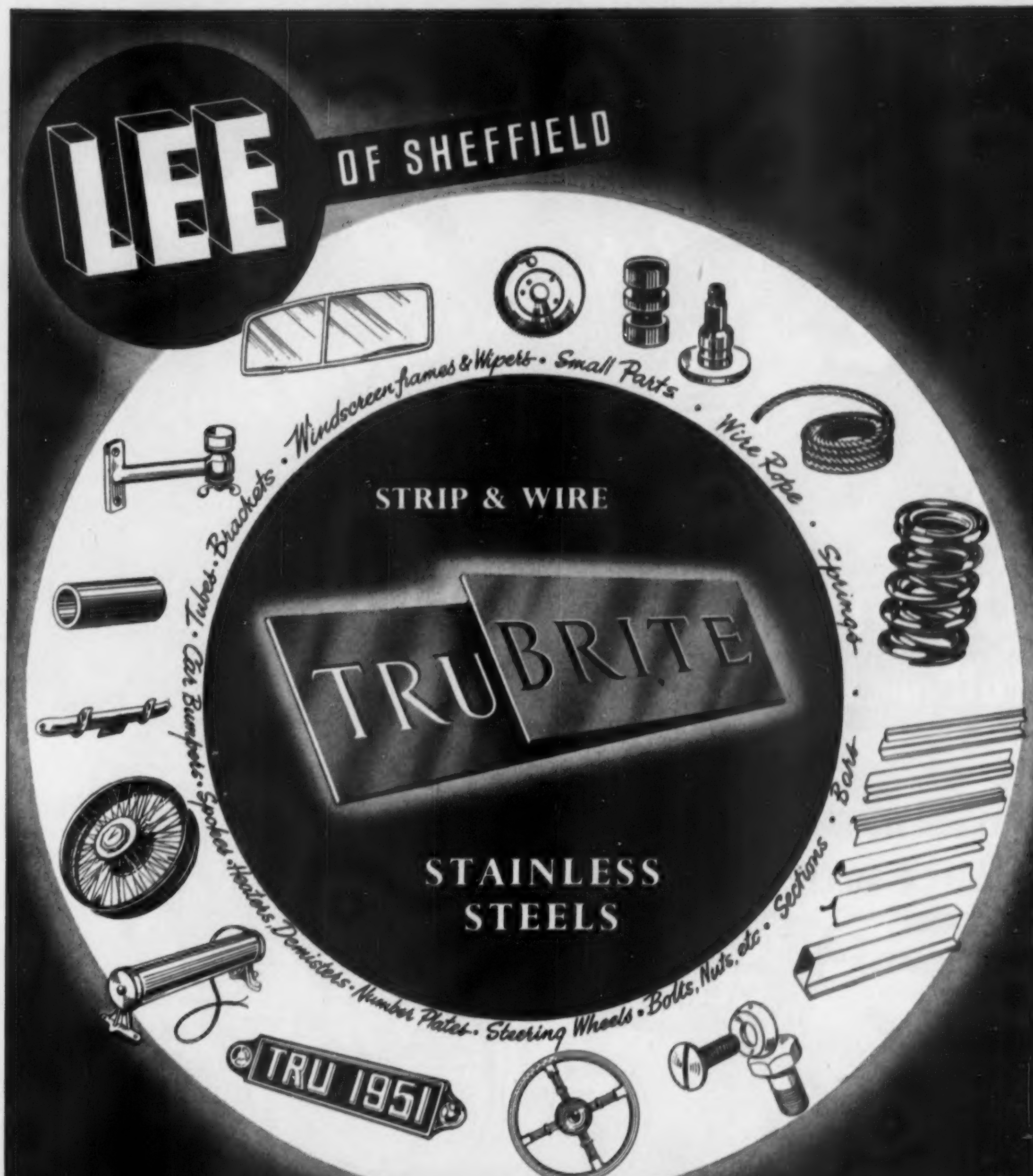

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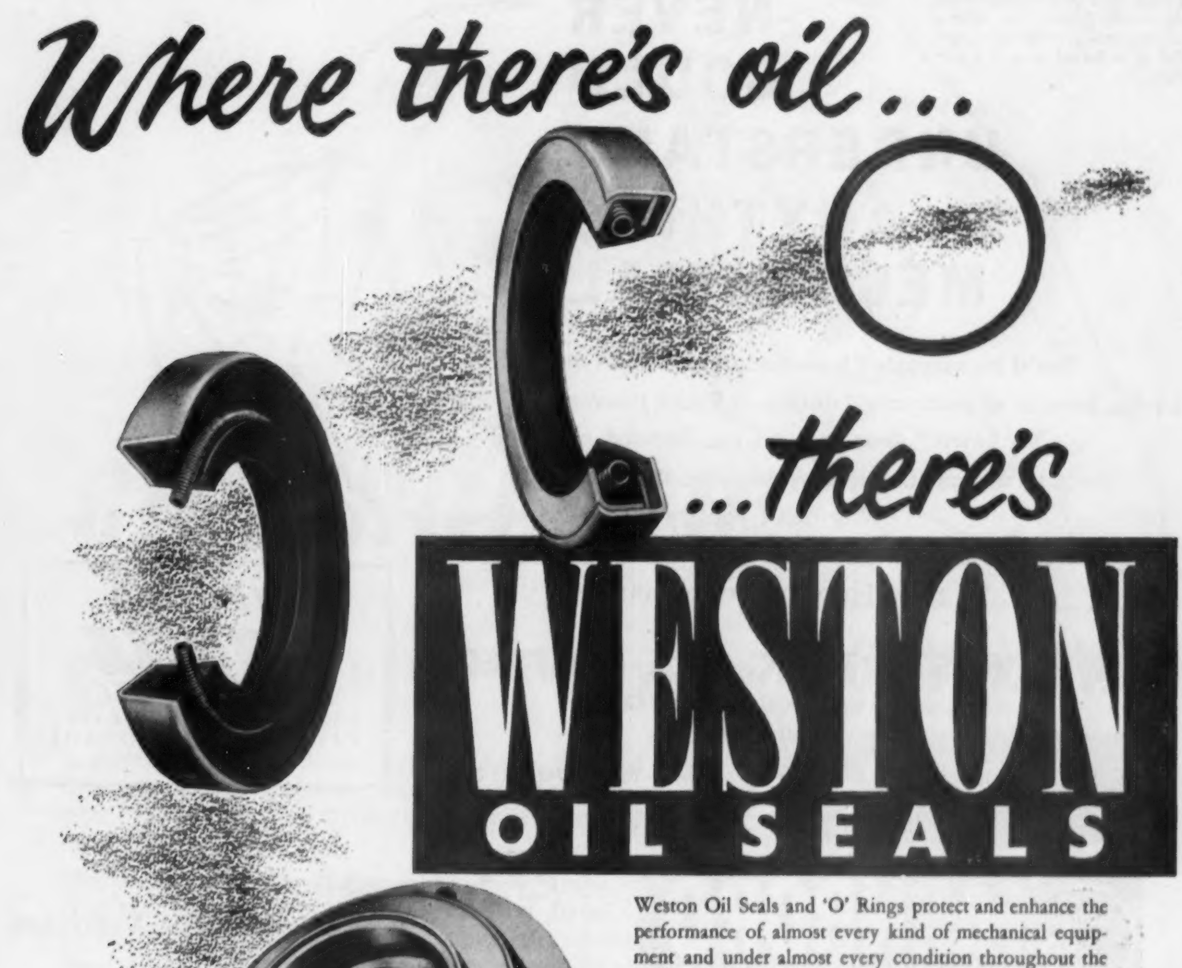
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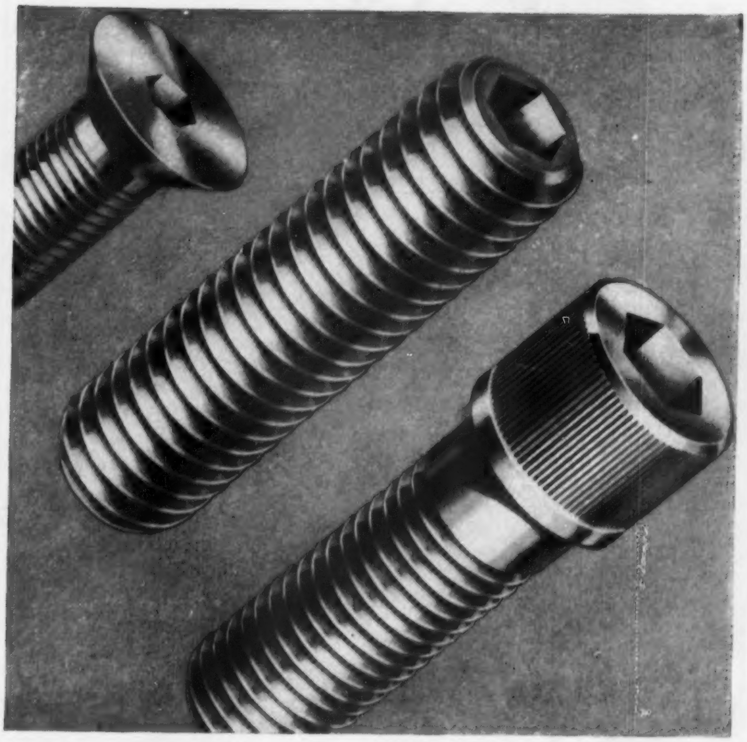

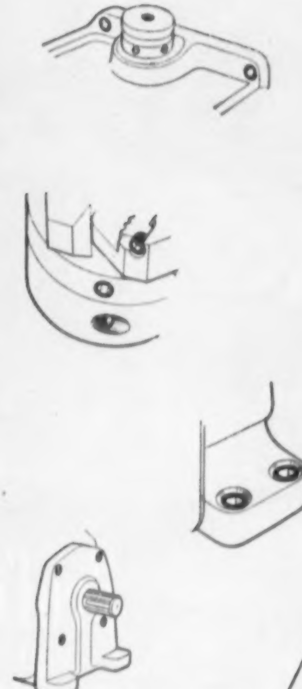

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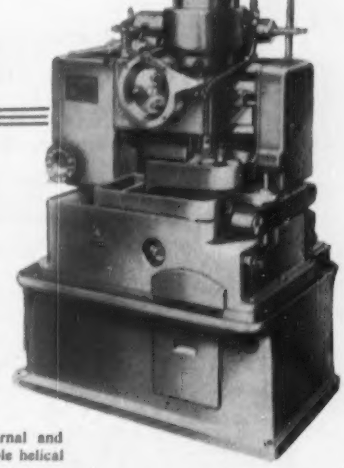

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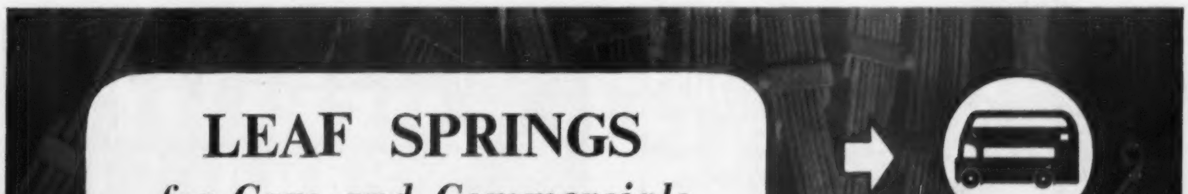
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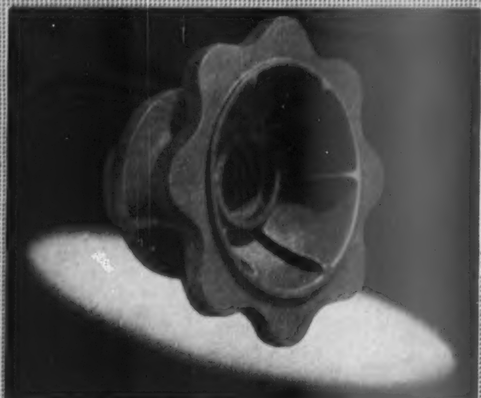

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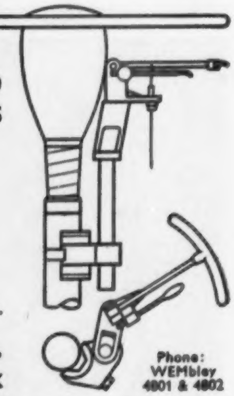

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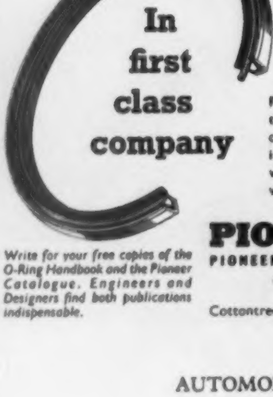

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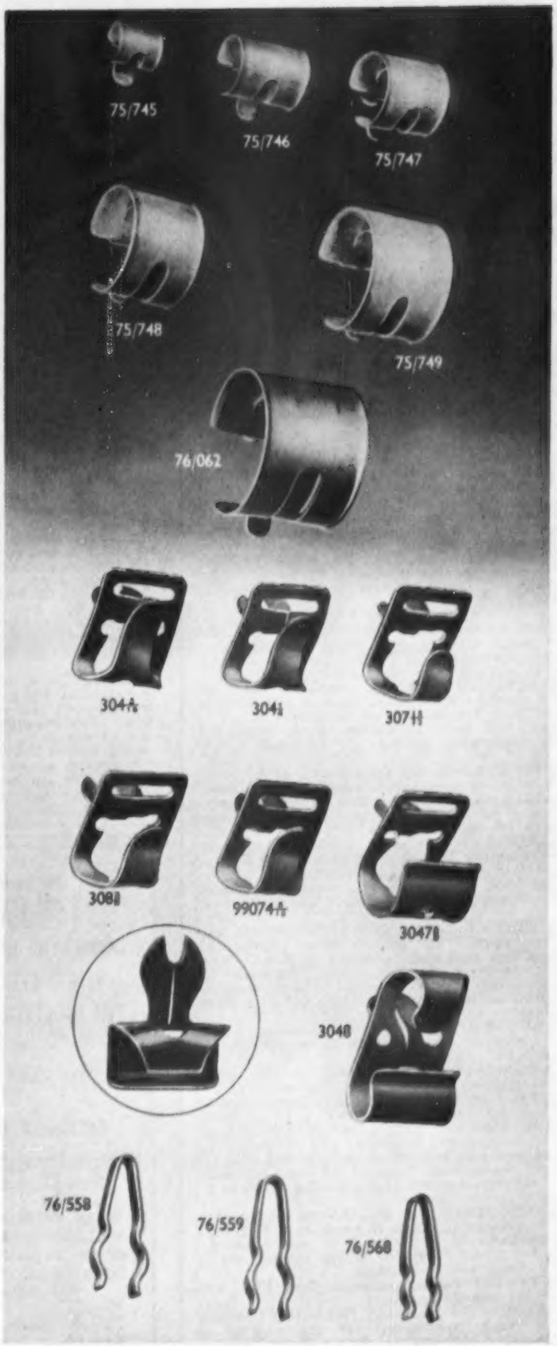

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